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"THE IMPORTANCE OF CONTROLLED HUMIDITY IN LONG TIME PRESERVATION."

BY COMMANDER GEORGE C. WELLS, U. S. N.

This paper was adjudged the best of those which were submitted for competition in the Society's Prize Essay Contest, 1948.

The Navy is placing primary reliance on maintenance of an atmosphere having comparatively low relative humidity for the preservation of interiors of ships which are retained as a part of the Navy but in inactive status. Nearly three years have elapsed since the undertaking of application of dehumidification to Reserve Fleet ships as a practical means of preventing their deterioration. The time which has elapsed permits that a reasonable evaluation of the adequacy of dehumidification for this purpose be made.

In the period between World War I and II, many advances were made in the development of preservative coatings and in

the techniques of their application. It was apparent, however, that preservative coatings alone would not insure the protection of the materials for indefinite periods of time. It was further apparent that for a vast number of components which go to make up a ship in a condition of material readiness, such that it could perform its designed functions without elaborate overhaul and costly reassembly at a time of activation, there was no practicable means of applying preservative coatings. Knowledge of the effect of atmospheric conditions on corrosion of metals, mold, mildew growth, and general deterioration of equipment indicated that control of relative humidity within the ship would result in ideal protection of all equipment located within the envelope. The Navy's Bureau of Ships, from the end of World War I until 1941, as the opportunity afforded, conducted tests with a view to determination of the practicability of such humidity control. Because of the indefinite need for specific information, however, not much formal effort along these lines was expended in this period. During this time, one company pioneered the idea and successfully accomplished the installation of machines and systems for the preservation of cargo by dehumidification. At the beginning of World War II, development efforts necessarily were diverted to higher priority items of active fleet equipment.

As the War progressed it became evident to the Bureau of Ships that upon termination of hostilities there would be huge quantities of materials which would require preservation at the lowest possible cost. Under direct leadership of Captain H. Gordon Donald, Captain Thomas H. Urdahl and Commander Elmer R. Queer, an intense program of research into the adequacy of dehumidification and into the development of practicable methods for its accomplishment was instituted. Resultant to accelerated tests, conclusions were drawn that:

(a) The maintenance of a relative humidity of 30 per cent would prevent deterioration from corrosion, mold, and mildew.

(b) This relative humidity was practicable of attainment within the envelope of a ship.

(c) That items of equipment and outfit could be allowed to remain aboard in their normal location, allowing the ship to serve as its own storehouse.

		Package	Large	Medium	Small
Adsorption Air CFM		20	500	250	150
Connected Load KW		1	8	4.6	3
Max. Cu. Ft. Volume of Shipboard Space Maintained at 30% RH		10000	400000	220000	150000
Adsorption Capacity pounds water per hour	95°F. 35% RH	0.225	9.4	4.7	2.9
	70°F. 35% RH	0.210	7.6	4.0	2.4

Table I - Capacity Characteristics of Dehumidifiers used in Reserve Fleet Preservation Program.

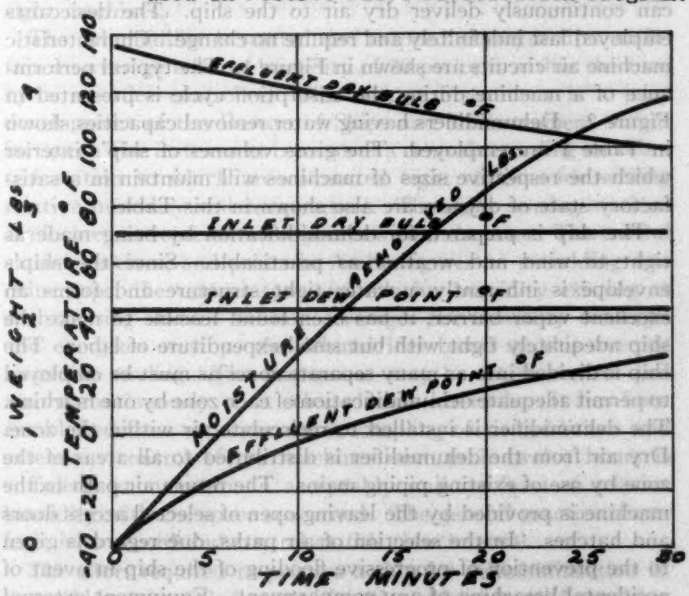

Figure 2 - Typical Performance Characteristics of a Large Dehumidifier, Adsorption Cycle. Entering Air at 70° F. 35% Relative Humidity.

(d) The distribution of dry air to permit attainment of uniform conditions of relative humidity was practicable of accomplishment.

Concurrent with research into the adequacy of principle, the Bureau of Ships, with the full cooperation and valuable advice of manufacturers, developed machinery suitable for the accomplishment of the job. Before the end of the War, necessary equipment characteristics had been established. With the cessation of hostilities, machines to perform the work were in production.

The dehumidifiers built employ solid desiccant as the drying agent. Energy for reactivating the saturated desiccant beds is electrical. A fan having sufficient available total pressure to circulate dry air through distribution mains is built integral with each dehumidifier. The machines are completely automatic in operation, working on a timed relationship between adsorption and reactivation cycle for the desiccant bed concerned. All but the smallest dehumidifiers in service are dual bed machines and can continuously deliver dry air to the ship. The desiccants employed last indefinitely and require no change. Characteristic machine air circuits are shown in Figure 1. The typical performance of a machine during the adsorption cycle is presented in Figure 2. Dehumidifiers having water removal capacities shown in Table 1 are employed. The gross volumes of ship's interior which the respective sizes of machines will maintain in a satisfactory state of dryness are also shown in this Table.

The ship is prepared for dehumidification by being made as tight to wind and weather as practicable. Since the ship's envelope is inherently a water tight structure and forms an excellent vapor barrier, it has been found feasible to make the ship adequately tight with but small expenditure of labor. The ship is divided into as many separate zones as must be employed to permit adequate dehumidification of each zone by one machine. The dehumidifier is installed to recirculate air within the zone. Dry air from the dehumidifier is distributed to all areas of the zone by use of existing piping mains. The return air path to the machine is provided by the leaving open of selected access doors and hatches. In the selection of air paths, due regard is given to the prevention of progressive flooding of the ship in event of accidental breaching of any compartment. Equipment external

to the ship's envelope, and which is subject to deterioration from effects of the atmosphere, is housed within inexpensive, weather and air tight, portable metal structures. The source of dry air to be circulated within these structures may be either the air within the ship's envelope or a low capacity automatic "package" dehumidifier. Selection of means of drying external equipment is made on the basis of the lower installation cost. Spaces within the envelope, through which it is found impracticable to circulate dry air, are kept dry by being opened to spaces where dry air is circulated. It will be noted from Table 1, that very small quantities of dry air relative to volume of the zone being dried are recirculated. A schematic diagram of typical ship's air circuiting is presented in Figure 3.

Numerically, the major proportion of ships under preservation are dried by a single machine. All ships of destroyer size and smaller have only one machine installed. Ships of cruiser size and larger auxiliary vessels usually require that three machines be installed. The largest combatant vessels require the use of eight machines.

The dehumidifier is started and stopped as required to maintain an average relative humidity of 30 per cent within the zone by a recorder controller. This unique instrument is a joint development of the Bureau of Ships and an instrument manufacturer. The instrument consists of a central recording and control station to which are connected eight stations for sensing relative humidity and temperature. The eight sensing stations are located at representative points within the zone. The central station receives, and prints in Arabic figures on an adding machine tape, the temperature and relative humidity existing at each sensing station location, averages the humidity readings, and energizes or stops the dehumidifier based on this average humidity reading. The instrument may be set to control and record continuously or at selected time intervals. It has been established from experience that sampling of air conditions and attendant control of the machines once each twelve hours is sufficient to maintain an acceptably uniform level of relative humidity with the space concerned. The general practice therefore, is to record and control at this time interval.

The effort required to prepare a ship for dehumidification is, as would be expected, directly a function of her material condition

at the time of inactivation. Preservative measures within the ship's envelope, preliminary to the application of dehumidification, consist of cleaning and drying of machinery and systems which in their functional nature contain water or are exposed to grease and contaminating oil vapors. Appropriate thin film rust preventative compounds are applied to the interior working surfaces of the machinery. These compounds have characteristics such that they need not be removed prior to use of the equipment. The thorough elimination of all free water inside the ship's envelope forms an important part of these steps. To be prepared for successful application of dehumidification, all under water leaks must be stopped. Leaks which permit the seepage of free water above the waterline as a result of rain fall, present far more troublesome sources of moisture than do openings to the weather which permit the entry of moisture laden air only. Ships of all-welded topside construction present fewer problems than do those of riveted construction. Experience has clearly shown that a condition of pressure tightness is not essential to the satisfactory operating economy of dehumidification equipment. It has equally demonstrated the fact that the equipment is operationally a costly bilge pump and that considerable care to prevent entry of free water is requisite to the successful application of dehumidification.

Many questions as to the efficacy of dehumidification may be raised. Some questions which may occur and the answers as they now appear to be evident may well be presented.

Is it indeed practicable to maintain and effectively control humidity aboard ship? Acceptable conditions are being successfully maintained on ships located in widely separated geographical areas. The ships require time varying from two weeks to two months in their initial drying period—that period of time during which residual moisture is being removed from materials within the envelope. After an equilibrium condition of 30 per cent is attained, it is successfully being maintained within a range of plus to minus five per cent from the average, using the methods of control earlier discussed. The scope of this successful application can be appreciated when it is realized that over three thousand machines are now in service.

Is humidity control costly of maintenance? The average maximum power requirements to date are five thousand kilowatt

hours per month per million cubic feet of dehumidified volume, the approximate volume of four ships of destroyer size. The average minimum requirements on vessels ideally suited for dehumidification are considerably less than half this requirement. In dollars and cents, assuming a power cost of $0.012 per kilowatt hour, a ship of destroyer size is maintained in a satisfactory condition of relative humidity at a power cost of approximately $180.00 a year. This indeed is a small price to pay for the preservation of so valuable an item of defense equipment as a ship of this type and size.

Is it essential to control humidity to insure long time preservation? Tests made by the Bureau of Ships and other interested Bureaus, Agencies, and Laboratories answer this question in the affirmative. No preservative material has yet been found which will maintain an item of equipment normally subject to deterioration by corrosion or mold in a state of readiness for immediate use and at the same time withstand the vicissitudes of cycling the equipment through the dry bulb temperatures at which condensation occurs. Items of equipment preserved with the best known preservatives and exposed to uncontrolled atmosphere have been observed to have deteriorated beyond the point of possibility of repair for use after as little as 90 day exposure. A group of vessels not belonging to the Navy, which, on being placed out of service, were laid up using preservative coatings alone, have recently been returned to service. These vessels required overhaul of virtually all operating components before they were usable. While costs may not be presented here, it may be stated that costs were many times those of placing in service similar ships which had been dehumidified for equal periods. The potential cost in man power chargeable to inadequate preservation which in the event of a required sudden mobilization and which might otherwise be applied to production in other fields is apparent.

Is it necessary that relative humidity of 30 per cent be maintained? As a corollary to this, is there a critical relative humidity below which deterioration will not occur and above which deterioration is rapid? To determine the answer to this and other questions, the Bureau of Ships established a test project in which a series of spaces are kept at fixed relative humidities in increments from 15 per cent to 90 per cent. To parallel condi-

tions which would exist aboard ship, dry bulb temperatures are not controlled. No effort is made to prevent the infiltration of dust or to control the presence of potentially corrosive gases other than water vapor since this would not be done aboard ship. In each of the spaces are placed samples of virtually all basic materials used in ship construction and in the ship's equipment and outfit. Additionally, such samples of operating equipment as can practicably be placed in the spaces are made a part of the test. Materials with and without preservative coatings are exposed. At regular intervals all items are inspected and where essential, are subjected to tests to determine physical properties and performance characteristics. The project has been active for more than three years and is continuing. A general view of the 90 per cent humidity test space is presented in Plate 1. The 30 per cent humidity test space is shown in Plate 2. The evident comparative general appearance should be noted. From evidence to date, it may be stated that with other conditions uncontrolled, there is no critical or break point of relative humidity. It may be further stated that a relative humidity of 30 per cent will so inhibit the formation of corrosion, mildew, and mold as to make possible the estimate that, for practical purposes, such formation is prevented. As humidity is increased to approximately 60 per cent relative, there is but small increase in the tendency toward deterioration by corrosion. There is, however, a more definite increase in the tendency of mildew and mold to form on animal and vegetable materials. At 90 per cent humidity, all materials subject to deterioration rapidly deteriorate with or without preservative coating. Plate 3 shows relative effects of stowage of a kapok life preserver in the two spaces. The comparison evident is characteristic of all vegetable and animal materials exposed. Plate 4 shows comparative effects on mild steel samples. It may be concluded that with other factors uncontrolled and for purposes of practicable application, a critical relative humidity does not exist, but that one of approximately 30 per cent will prevent deterioration.

What happens when corrosion is active at the time dehumidification is applied? Evidence to date reveals that, at 30 per cent humidity, corrosion stops. As time passes, the rust film

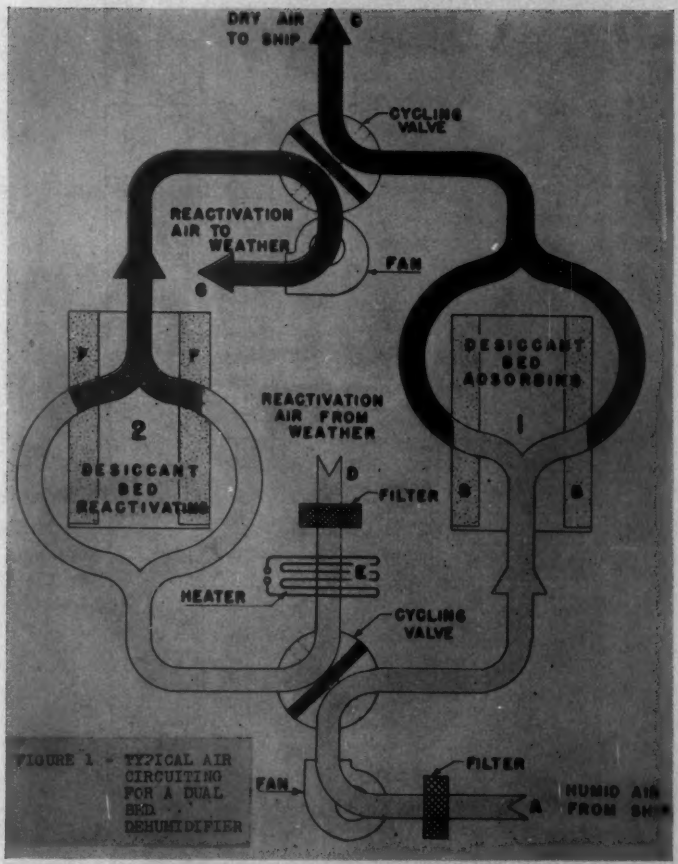

FIGURE 1 - TYPICAL AIR CIRCUITING FOR A DUAL BED DEHUMIDIFIER

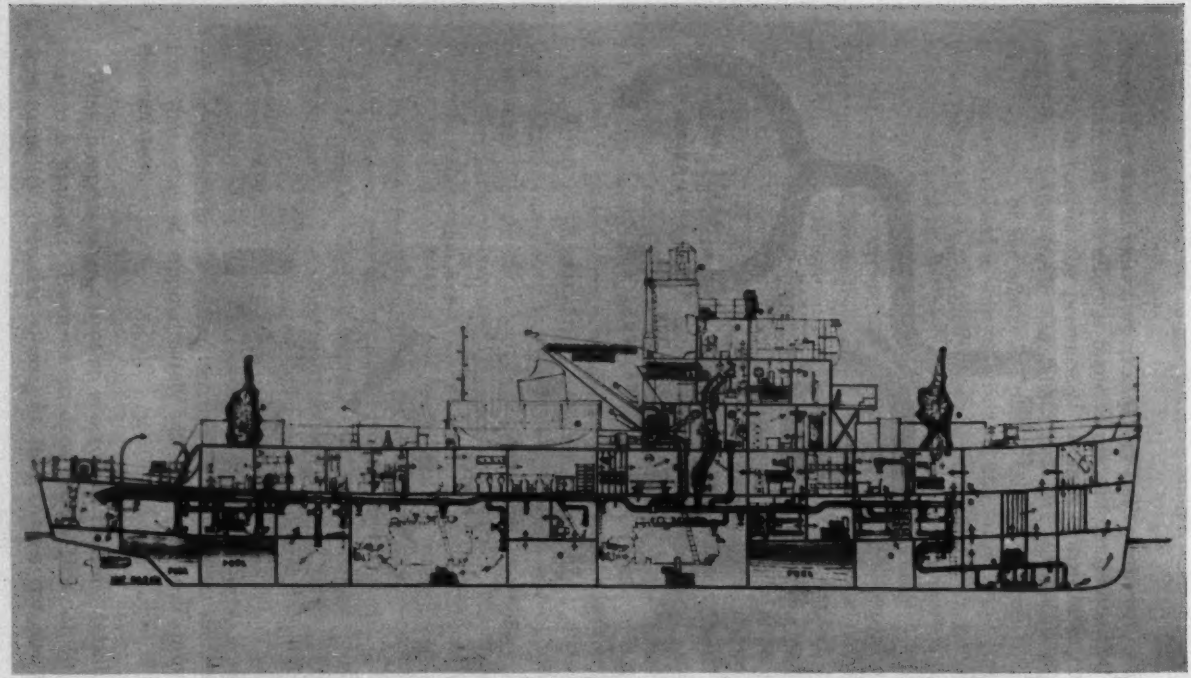

FIGURE 3.—TYPICAL DRY AIR DISTRIBUTION SYSTEM FOR A DYNAMICALLY DEHUMIDIFIED SHIP. NOTE THAT DRY AIR FROM MACHINE IS DISTRIBUTED USUALLY THROUGH EXISTING PIPING MAINS AND THAT HUMID AIR RETURN PATH TO MACHINE IS BY WAY OF OPEN ACCESS DOORS AND HATCHES.

PLATE I GENERAL VIEW OF TEST CABINET AFTER 3 YEARS EXPOSURE TO ATMOSPHERE OF

PLATE II GENERAL VIEW OF TEST CABINET AFTER 9 YEARS EXPOSURE TO ATMOSPHERE OF

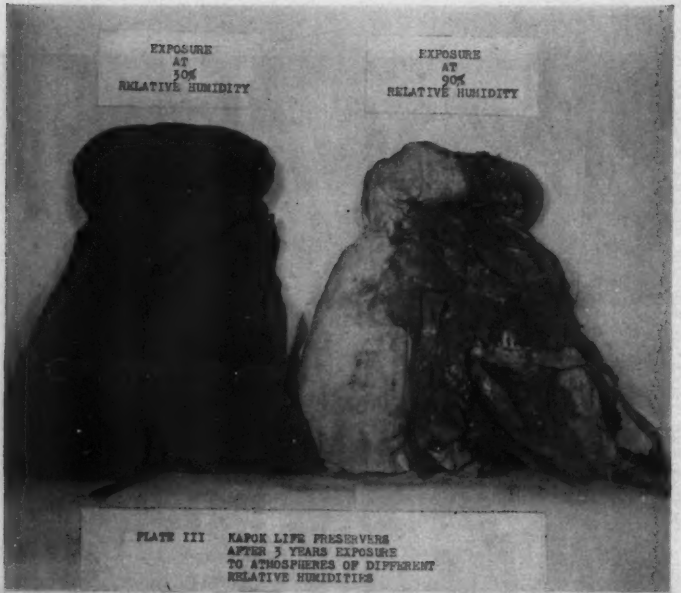

PLATE III KAPOK LIFE PRESERVERS
AFTER 3 YEARS EXPOSURE
TO ATMOSPHERES OF DIFFERENT
RELATIVE HUMIDITIES

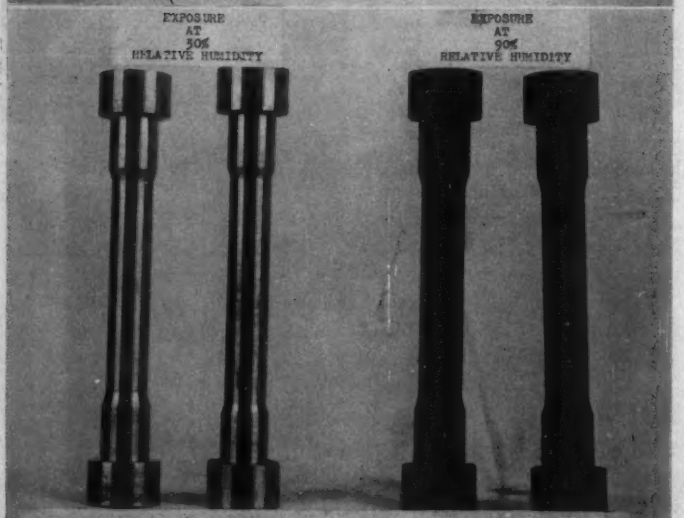

PLATE IV MILD STEEL SPECIMENS AFTER 3 YEARS EXPOSURE
TO DIFFERENT RELATIVE HUMIDITIES

becomes so dry that the light powdery residue may be brushed off with ease.

Since the deterioration of materials is prevented by maintaining an atmosphere of low relative humidity, why is it essential that preservative coatings be used at all? If equipment is opened up to permit that it rapidly come in equilibrium with the 30 per cent atmosphere, manifestly that equipment is not ready for immediate operation. Further it is not practicable to open some equipment at all. If not opened up, the time for complex interior surfaces to reach an equilibrium of 30 per cent relative humidity is considerable. This period required for machinery to reach equilibrium conditions inside and out is sufficiently long that the item in question will deteriorate past the point of operability unless additional preservation measures are taken. For equipment essentially open to the atmosphere, no preservative measures additional to the control of relative humidity are necessary.

It having been established that the use of preservative coatings additional to the maintenance of a dehumidified atmosphere is essential under certain circumstances and the reason for this being that just presented, is it established that equipment of this nature is in fact adequately preserved for periods indefinitely long? As a part of the test project mentioned earlier, various complex units of machinery were inactivated, using preservative films of such nature that they did not impair the immediate readiness of the equipment to operate without removal of preservatives. The items were then placed in an atmosphere of 30 per cent humidity. At intervals, varying from six months to three years of no further attention, these equipments have been started and found to function as designed. They have then been torn down and minutely inspected for evidence of deterioration. None has been observed. Similarly, the trial reactivation of a limited number of dehumidified ships has revealed that all equipment functioned normally where cleaning and necessary application of preservative films had been accomplished at time of inactivation. Additionally, detailed inspection of representative items of equipment on board ships inactivated for a period from six months to over two years have revealed that the items have not deteriorated. It may be concluded that the equipment under consideration is adequately preserved.

The ships are for practicable purposes sealed against entry of air. Is there not a probability that oxygen depletion or formation of toxic or explosive gases will make the ships dangerous to enter for necessary inspections? These factors have been carefully observed on representative ships during the time they have been in inactive status. All evidence to date points to the fact that there is absolutely no cause for concern in the premises. On the contrary, the desiccant beds of the dehumidifiers adsorb many vapors other than water on the drying cycle and release them to the weather on the reactivation cycle. There is, if anything, a tendency to reduce the quantities of objectionable vapors which might initially be present in the ship.

Since many materials normally attractive to vermin and other forms of animal life are retained on board, might it not be expected that some inactive ships will become infested and that these materials will be destroyed? While not so immediate in its effect as fumigation, dehumidification is just as deadly to this animal life over the long period. There simply is no water on which the animal may subsist. As a consequence, the dehumidified ship is entirely free of vermin and none seek to get aboard. This fact has been spectacularly apparent on some ships.

Might not spaces well be dehumidified by static means instead of by resorting to automatic machinery? The average water removed per month from a zone of a typical surface ship has been approximately 525 pounds. The zone has a volume of 150,000 cubic feet. Assuming that the statically used desiccant is to be changed once each month, and assuming the use of materials of high adsorptive capacity, the charge of desiccant for the zone would have to be approximately 2625 pounds. This desiccant necessarily must be distributed throughout the zone. For a frequency of change established at one month, the capital investment in desiccant material alone approximates that of the automatic dehumidifiers in use. If the frequency of change were to be at longer intervals, this investment would soon exceed the cost of automatic equipment. Labor required to effect the changes of a statically used drying agent is to be compared to the approximate total of one man day per month required for the routine maintenance of the automatic equipment installed and for the inspection and recording of conditions within the zone. The additional moisture required to be removed from the

zone incident to its opening to remove and replace desiccant has not been considered in this comparison. This load would materially increase the static charge required over that here shown to be essential. Power costs for reactivating would equal or exceed those for reactivating desiccants within the dehumidifiers since the machines operate at a high efficiency in this respect. It is apparent, that on a comparative basis, the cost of static dehumidification is prohibitively high. Static dehumidification provides a valuable and efficacious means of limiting moisture vapor content of the air when the container is hermetically sealed and is a perfect moisture vapor barrier. These requisite conditions are not generally encountered aboard ship. Static dehumidification is used to a limited extent in the Reserve Fleet program. Its application is limited to spaces which may be sealed and which do not lend themselves to incorporation in the dry air distribution system from the installed machines.

Finally, is it not possible that the low relative humidity conditions will of themselves directly destroy certain materials? All basic materials have been carefully observed with this in mind. No deterioration of any material attributable to low humidity has been observed. Certain materials, after reaching equilibrium at very low humidities, must be allowed to regain their normal moisture content before they show normal properties. This regain is very rapid. It should be borne in mind that 30 per cent is not an extremely low humidity, but is one normally encountered in heated office spaces and homes during the winter season.

The general appearance of a ship laid up in accordance with the principles herein presented is impressively excellent. The interiors parallel in appearance those of a ship just delivered from the builder and awaiting the appearance of her crew. The odor is that of a clean fresh atmosphere, having no remote relationship to the musty odor ordinarily associated with spaces long unused. The appearance has been described as that of a ship whose crew has just left for light refreshments and which was scheduled to and capable of getting underway on a moment's notice. A most careful detailed inspection of components reveals that this general appearance is not deceptive. The equipment is found to be in the same condition in which it was left many months before, and in a state of complete readiness for use. The

author was intimately associated with the reactivation of a number of destroyers and submarines of the "Red Lead Fleet," just prior to this Nation's entry into the recently concluded hostilities. His most vivid recollection of that period is associated with the fact that for many months his epidermis was covered with a mixture of grease and rust that defied removal by the most powerful of solvents. A comparison of Plates 1 and 2 is recommended as being representative of the material conditions encountered in that earlier reactivation and a reactivation of a dehumidified ship which might in the future be required. As of this date, it must be concluded that the principles of preservation established for the maintenance of the Reserve Fleet are sound and that dehumidification provides an eminently satisfactory solution to the problem of long time maintenance of ships in an excellent state of preservation and immediate readiness.

THE USE OF THERMOCOUPLES IN HIGH-VELOCITY GAS STREAMS.

By Ernest F. Fiock[1] and Andrew I. Dahl[1].

Research Narrative.

The general purpose of this paper is to present background information which may be useful to those confronted with the problem of determining, with a practicable minimum of error, the temperature of a stream of hot gases such as may be encountered in a modern steam or gas turbine. Since the thermocouple is at present the most satisfactory and widely used instrument for this application, the discussion is limited to such temperature-sensing instruments.

This report reviews the methods which may be used and the precautions which must be observed in obtaining precise and reliable gas temperatures, and includes general instructions for choosing the thermoelements, making the measuring junction, terminals, leads, reference junction and reference-junction bath, for selecting the measuring instrument, and for calibrating the wire and applying the calibration data.

The individual effects of thermal radiation and conduction, of the transfer of heat to the measuring junction by convection, and of gas velocity are discussed by reference to equations and charts. By means of specific examples, some of these effects are related to the size, shape and material of the thermocouple or thermocouple well. It is shown that conduction along solid parts is of little consequence in most practical installations; that radiation is also of little importance in steam systems with well lagged pipes, while the velocity effect becomes increasingly important as the development of power plants progresses; and that radiation is the most serious problem in existing gas turbines and jet engines.

Since heat is transmitted to a measuring junction immersed in a stream of flowing gas by direct impact, it has been found desirable to speak of a static and of a total or dynamic temperature of the gas. The difference is a function only of the velocity and heat capacity of

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the gas, and is seldom of practical significance at velocities below 300 ft per sec. Its magnitude increases as the square of the velocity, so that it assumes considerable importance at flow rates now prevailing in some power plants.

It is pointed out that the static temperature is not subject to direct measurement, that the total temperature can be measured with an instrument of a type which is not suitable for installation in many power plants, and that it is possible to determine experimentally the correction which must be applied to convert the indicated temperature to either static or total temperature. Such correction involves the use of a recovery factor which can be measured for the particular junction or thermocouple well being used in the gas stream. The recovery factors of various junctions and thermocouple wells are being measured in air at the National Bureau of Standards and in steam at the Naval Boiler and Turbine Laboratory, Philadelphia, under sponsorship of the Bureau of Ships.

INTRODUCTION.

An experimental determination of the thermal efficiency of a modern steam or gas turbine consists essentially of measurements which establish the state of the working medium at chosen locations within the power plant. The state of a gaseous medium in turn is fixed if its composition, pressure and temperature are known, since from these the other thermal properties may be found in existing tables. This paper reviews the precautions which must be observed in order that the experimental error in measuring the temperature of a gas stream with a thermocouple may be kept to a practicable minimum. These precautions have been established through the efforts of numerous investigators over a period of years, and are assembled here particularly for those faced with the problem of measuring gas temperatures with high accuracy in the somewhat unfamiliar range encountered in recent steam power plants and in gas turbines.

Since a determination of thermal efficiency with a given accuracy requires that the enthalpy of the working medium be known with at least this same accuracy, it is a simple matter to determine from tables of the thermal properties of the working medium the corresponding tolerance in the observed value of temperature. Typical examples of the maximum allowable error in temperature for an error of $\pm 0.1\%$ in enthalpy are: For superheated steam

at 500 lb per sq in. and 500°F, ±1.6°F; for superheated steam at 500 lb per sq in. and 1000°F, ±2.8°F; for air at low pressure and 1000° and 2000°F, ±1.0°F and ±1.9°F, respectively.

NOMENCLATURE.

The nomenclature used throughout this report is as follows:

Symbols.

emf—thermal electromotive force
D—diameter
R—effective radius = $1/2\sqrt{D^2 - D_i^2}$
a—depth of immersion
k—thermal conductivity
t—temperature in degrees Fahrenheit
T—temperature in degrees Rankine
h—coefficient of heat transfer by convection
C_p—specific heat at constant pressure
γ—ratio of specific heat at constant pressure to that at constant volume
G—mass rate of flow per unit cross-sectional area
μ—absolute viscosity
σ—Stefan-Boltzmann radiation constant
ϵ—surface emissivity of measuring junction
r—recovery factor
u—mean velocity of random molecular motion
V—directed velocity of mass motion
p—absolute pressure
ρ—gas density

Subscripts

g—gas
w—wall
j—junction
f—at the prevailing state
i—inside, applied to D; indicated, applied to t or T
s—static
t—total

APPARATUS, METHODS AND PRECAUTIONS NECESSARY FOR HIGHEST ACCURACY UNDER ALL OPERATING CONDITIONS.

A thermocouple (1)* is essentially "a pair of electrical conductors so joined as to produce a thermal electromotive force

* Numbers in parentheses refer to similarly numbered references in bibliography at end of paper.

(emf) when the junctions are at different temperatures." One junction, called the reference junction, is maintained at a known temperature, as for example at 32°F in a mixture of water and melting ice. The second junction, called the measuring junction, is affixed to or immersed in the medium whose temperature is to be observed. The emf is measured with a millivoltmeter or a potentiometer.

When high accuracy is required, two things must be accomplished: The actual temperature of the measuring junction must be determined; and the difference in temperature between the measuring junction and the medium must be known. Under special conditions suitable means may be found for making the latter negligibly small. If this is not possible, as is often the case, the magnitude of the difference may be determined by experiment or from calculations, if the conditions of the surroundings are known.

Equipment Required for a Thermoelectric Measurement.

The various elements which are basic in the application of thermocouples at high temperature are well established (1), but are reviewed briefly for the sake of completeness.

Choice of Thermoelements. Comparatively few combinations of metals and alloys have been found useful for thermocouples, so that the choice of elements appropriate for a particular application is usually quite simple. The combinations used most frequently (See Table I of reference 1) are listed below, together with the temperature to which each is serviceable and the average thermoelectric power over a specified range of temperature.

Thermoelements	*Usual Temperature Limit °F*	*Average EMF per °F in the Range Specified microvolts/°F*
Copper-Constantan (Cu-Kn)	650	28 (32-650°F)
Iron-Constantan (Fe-Kn)	1400	32 (32-1400°F)
Chromel-Alumel (Ch-Alu)	2200	23 (32-2200°F)
Platinum-Platinum Rhodium (Pt-90%Pt, 10%Rh)	2650	6.3 (1000-2650°F)
Chromel-Constantan (Ch-Kn)	1400	42 (32-1400°F)

Of the above materials, copper, iron and constantan oxidize more rapidly at a given high temperature (2) than the others.

Chromel and alumel form protective oxide coatings which make them highly resistant to further oxidation. The combination Ch-Kn has a very high thermoelectric power, but its service life is limited to that of the Kn. Except for a rapid alloying with metallic silicon, which may be produced if ceramic insulating tubes are used in a reducing atmosphere, both Pt and PtRh are chemically inert, but the platinum metals are well known for their ability to act as catalysts. For this latter reason the use of Pt-PtRh in exhaust gas below 1200°F should be avoided. Although there is a natural tendency to maintain a Pt-PtRh thermocouple as a comparison standard against which other couples are checked, this is poor practice in the range below 1000°F because of the relatively low thermoelectric power of the noble-metal couple. In this range platinum resistance thermometers or calibrated base-metal thermocouples are preferable.

Making the Measuring Junction. The measuring junction may be made either with solder having an appropriate melting point or by autogenous welding. If solder is used the quantity should be kept to a practicable minimum, and if flux is used in either method the excess should be removed in boiling water. Low-melting solders, such as alloys of lead and tin, flow freely on Cu, Kn, and Fe, but not on Ch or Alu. There seems little point in using any type of solder on noble-metal couples, since these weld so easily in either an arc or a flame. High-melting solders, such as those containing a high percentage of silver, may be used on Cu, Kn, Fe, Ch and Alu. If the junction is made without twisting the wires together, there can be no doubt as to its exact location.

Lead Wires and Terminals. Where highest accuracy is sought it is preferable that the thermoelements extend without a break from the measuring to the reference junction. In large power plants this is not always practicable, and the elements must be broken at binding posts in the head of the instrument. In this case the lead wires should have the same thermoelectric properties as the elements, and preferably should be from the same spool of wire. If, as is usual, there are temperature gradients in the binding posts, the electrical circuit between such thermoelement and its lead should be completed through the same alloy as the element. Such an arrangement, shown in Figure 1, eliminates spurious thermal emfs which arise at the junctures of dissimilar

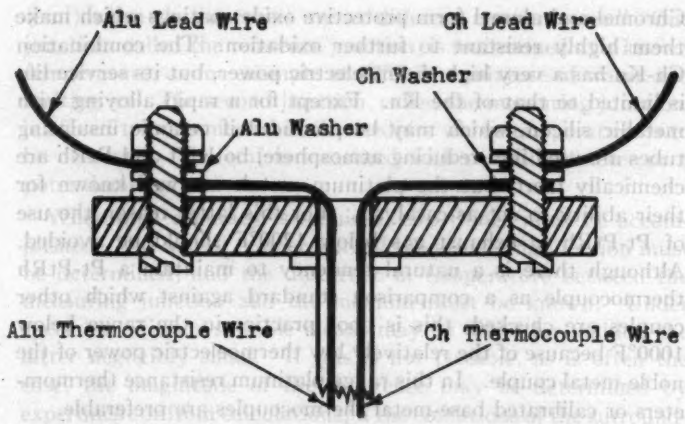


FIGURE 1.—TERMINALS FREE FROM STRAY THERMAL ELECTROMOTIVE FORCES.

metals in the presence of a temperature gradient. The leads, particularly if they are in the vicinity of power lines or electrical equipment, should be within a grounded metal shield, such as a pipe or duct.

Temperature of the Reference Junction. An intimate mixture of shaved ice and water provides the most convenient and easily reproducible bath for the reference junction. A small glass tube dipping into the slush, and containing a few drops of mercury, affords a convenient means of making the electrical connection between each thermocouple lead wire and the copper wire to the measuring instrument. These two wires should be insulated from one another, except below the mercury. The mercury should be immersed several inches in slush, so that heat conducted down the wires and the glass tube does not reach the junction. Shaved ice should be added as needed to insure that the bottom of the tube is never surrounded by water alone.

When an ice bath can not be used, a thermally-insulated copper block or box, or a stirred liquid bath may be substituted. In these cases the temperature of the reference junction must be measured with an auxiliary instrument.

Many indicating and recording potentiometers have built-in means for eliminating the need for a controlled reference bath.

Such devices, called reference-junction compensators, are convenient and satisfactory for work of moderate accuracy, but should not be used when the highest accuracy is required.

Selector Switch. If a single indicator or recorder is used for several thermocouples, a selector switch may be used. The copper leads from the reference junctions are connected to this switch and copper leads run from the switch to the instrument. No metal other than copper should form a part of the electrical circuit beyond the reference junction unless all regions of contact between dissimilar metals are kept free of temperature gradients.

Indicating or Recording Potentiometer. For highest accuracy the use of recorders should be avoided and a precision or a semi-precision potentiometer, rather than a millivoltmeter or a recorder, should be used. Obviously the potentiometer and its standard cell should be calibrated, and the instrument must be kept in proper balance during use.

Interpretation of Thermocouple EMF in Terms of Temperature.

Specifications for Thermocouple Wire. After a decision is reached as to the size and combination of thermoelements best suited to a particular need, it is desirable to procure a supply of these elements in a single quantity ample for all predictable applications on a given job. It is good economy to procure the wire under specifications appropriate to the need and covering the points made in the following specific example: Five hundred feet of No. 20 gage chromel-alumel thermocouple wire, duplex, the wires insulated individually with glass braid, and with a braided glass wrapper overall; each thermoelement to be in one continuous length; the emf-temperature relation of the wire must conform to that given in Table 2 of National Bureau of Standards Research Paper No. 767 (3) within ± 5°F up to 660°F and within $\pm 3/4\%$ of the temperature in °F at temperatures above 660°F; the thermocouple wire shall be given a stabilizing heat treatment.

If a great length of wire is needed, the requirement that the individual wires be continuous lengths may be waived for an alternate specification that all wire must be from the same batch.

Suggested tolerances for deviations from appropriate reference tables for other thermocouples are as follows:

Couple	*Reference Table*	*Tolerances*
Cu-Kn	Table 8, RP1080 (4)	±4°F to 650°F
Fe-Kn	Table 10, RP1080 (4)	±5°F to 500°F; ±1% above 500°F
Ch-Alu	Table 2, RP767 (3)	±5°F to 660°F; ±3/4% above 660°F
Pt-Pt10%Rh	Table 4, RP530 (5)	±3°F to 2200°F

Calibration of the Wire. For accurate results calibration of the wire is always required, the number of calibrations depending upon the uniformity of the wire. This may be gaged by calibrating couples made from samples cut from each end and from the center of the spool to determine how the thermoelectric properties vary throughout the stock of wire. If these variations are small as compared with the allowable error in temperature, no further calibration is necessary. Frequently the variation throughout a single spool is too great to permit such a simplified calibration, and it is necessary to calibrate wire immediately adjacent to that actually used for each measuring junction.

If the user has the necessary equipment and experience, calibrations may be made in his own laboratory. If this is not the case he may choose to have the calibration done at a laboratory which specializes in such work. The methods of calibration in use at the National Bureau of Standards are described in reference (6).

From the foregoing discussion it will be apparent that care must be exercised in maintaining the identity of each measuring junction, so that proper leads and corrections may be used therewith.

General Precautions in the Use of Thermocouples at High Temperatures.

Some of the base-metal alloys used as thermoelements for high-temperature service undergo gradual changes, probably in structure, upon extended exposure to high temperatures. The stabilizing heat treatment mentioned in a previous section helps to bring about such changes prior to the calibration of the wire, but can not do so completely.

When an inhomogeneous portion of a thermocouple is in a temperature gradient, a spurious thermal emf will result. Depending upon the prevailing conditions this might add to or

subtract from the emf of the measuring junction and introduce an error for which correction is impossible. Thus a junction calibrated at high temperature should not be used as a measuring junction in service, but rather the latter should be made from adjoining wire on the spools. Once a thermocouple has been hot for some time, it should not be used thereafter with less immersion, since a region in which changes in structure are probable would then be subjected to a large temperature gradient. For the same reason, a used thermocouple should not be recalibrated unless this can be done in place so that the portion subjected to the gradient is not changed.

Since base-metal thermoelements are very inexpensive, frequent replacement of thermocouples used at high temperatures is good economy. If enough of the original thermocouple is available, that portion which has been hot may be removed and a new working junction made, without a recalibration.

Use of the Calibration Data. Standard tables of emf vs. temperature may be found in the literature for all of the combinations which are in general use. It is convenient to express the results of a calibration in terms of deviations from a specified reference table, a curve of such deviations vs. observed emf being commonly used. Thus, to determine the temperature corresponding to an observed emf, the correction at this observed emf is obtained from the deviation curve. This correction is applied and the corresponding temperature is then read from the reference table.

The Difference in Temperature Between the Measuring Junction and the Medium of Unknown Temperature.

Since only the temperature of the junction and not that of the medium can be measured directly, it is obvious that the difference in these two temperatures must be either evaluated or reduced to a negligible amount. Since such evaluation involves many uncertainties, it is often expedient to employ other means for reducing the difference to the smallest value practicable.

Referring to a junction within a pipe through which hot gas is flowing slowly enough that the effect of velocity on temperature is negligible, it will be apparent that for moderate gas temperatures the wall temperature can be made to approach that of the gas by the use of lagging. When such a condition

prevails, as it does in many steam systems, even a bare junction will attain essentially the same temperature as the gas. Thus the temperature of steam in existing power plants can be measured with thermocouples without correcting for heat losses by conduction and radiation, provided the pipe is lagged adequately, and has reached a steady state.

For higher gas temperatures, such as those encountered in gas turbines, it becomes increasingly difficult and eventually impracticable to bring the pipe-wall temperature to equality with that of the gas by lagging. This, plus the fact that the rate of heat exchange by radiation increases as the fourth power of the absolute temperature, indicates that losses from the measuring junction by radiation and conduction must be accounted for if thermocouples are to give accurate values of the temperature of the working medium in gas turbines.

In both steam and gas turbines account must be taken of the effect on thermocouples of gas velocity if the latter is high. This correction will be discussed in detail in a succeeding section.

The individual effects of conduction and radiation upon the temperature of the junction will be considered next. To avoid confusion as to the direction of heat flow, it will be assumed that the wall is colder than the gas, since this is the condition which prevails in power plants. The same analysis of the problem applies, however, if the reverse is true.

Loss by Direction Conduction. Consider, as indicated in Figure 2, that a tube (or a thermocouple wire) of external diameter D, internal diameter D_i, and effective radius $R = 1/2\sqrt{D^2 - D_i^2}$ (or $R = 1/2\ D$ for a wire) is immersed for a distance a in a gas of temperature t_g and is conducting heat to a wall having a lower temperature t_w. Let t_j be the temperature of the tube at $x = 0$, t be the temperature at x, and consider the heat flow to and from the element δx at a steady state.

The quantity of heat entering δx at x in unit time is $-\pi R^2 k(dt/dx)$, and that leaving δx at $x + dx$ is $-\pi R^2 k[dt/dx + \delta x(d^2t/dx^2)]$. The quantity of heat flowing from the gas to the element δx is $-\pi D \delta x h(t - t_g)$. Since a steady state exists, the element δx must receive and lose heat at equal rates, so that

$$D\delta x h(t - t_g) + R^2 k(dt/dx) = R^2 k[dt/dx + \delta x(d^2t/dx^2)].$$

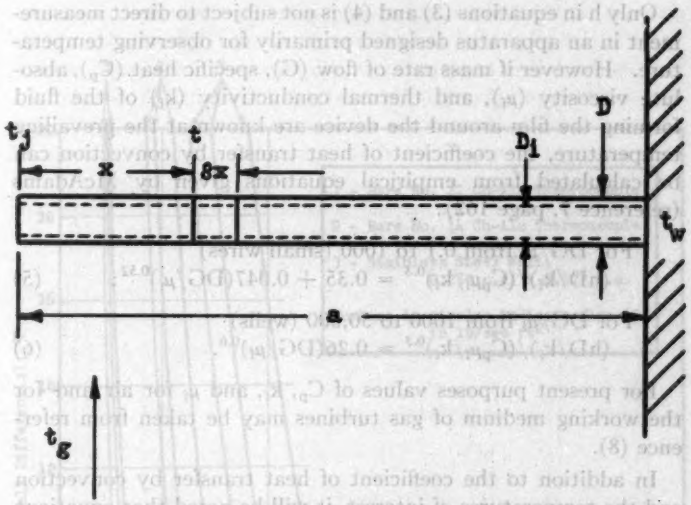


FIGURE 2.—NOTATION USED IN DERIVING THE EXPRESSION FOR HEAT LOSS BY CONDUCTION.

To simplify the notation, let $\alpha^2 = Dh/R^2k$, and the above equation reduces to

$$d^2t/dx^2 = \alpha^2(t - t_g). \tag{1}$$

The solution of this equation is

$$t - t_g = Ae^{\alpha x} + Be^{-\alpha x}, \tag{2}$$

in which A and B are constants which may be evaluated from the boundary conditions that $dt/dx = 0$ at $x = 0$ and $t = t_w$ at $x = a$. The final result is

$$\frac{t_g - t}{t_g - t_w} = \frac{\cosh \alpha x}{\cosh \alpha a} = \frac{\cosh x\sqrt{Dh/R^2k}}{\cosh a\sqrt{Dh/R^2k}} \tag{3}$$

For the special case of a thermocouple wire leading to a junction at $x = 0$, the temperature of the junction may be represented by t_j and $R^2 = D^2/4$, so that equation (3) reduces to

$$\frac{t_g - t_j}{t_g - t_w} = \frac{1}{\cosh 2a\sqrt{h/Dk}}. \tag{4}$$

Only h in equations (3) and (4) is not subject to direct measurement in an apparatus designed primarily for observing temperature. However if mass rate of flow (G), specific heat (C_p), absolute viscosity (μ_f), and thermal conductivity (k_f) of the fluid forming the film around the device are known at the prevailing temperature, the coefficient of heat transfer by convection can be calculated from empirical equations given by McAdams (reference 7, page 162):

For DG/μ_f from 0.1 to 1000 (small wires)

$$(hD/k_f)/(C_p\mu_f/k_f)^{0.3} = 0.35 + 0.047(DG/\mu_f)^{0.52}; \quad (5)$$

For DG/μ_f from 1000 to 50,000 (wells)

$$(hD/k_f)/(C_p\mu_f/k_f)^{0.3} = 0.26(DG/\mu_f)^{0.6}. \quad (6)$$

For present purposes values of C_p, k_f, and μ_f for air and for the working medium of gas turbines may be taken from reference (8).

In addition to the coefficient of heat transfer by convection and the temperatures of interest, it will be noted that equations (3) and (4) contain the depth of immersion a, the thermal conductivity k, and the shape parameters D and R. It is therefore difficult to present tables or charts which show the individual effects of each of these factors on the actual temperature assumed by the measuring junction.

One useful tool is a plot of the ratio $(t_g - t_j)/(t_g - t_w)$ vs. a^2h/kD, as shown in Figure 6 of reference 9. For any specific installation and assumed flow condition, h can be calculated from equations (5) or (6), after which the ratio a^2h/kD can be computed. The observed values of t_j and t_w, in conjunction with the value of the ratio $(t_g - t_j)/(t_g - t_w)$ read from the curve, permit the calculation of the desired value of t_g. Graphical methods for making such computations are presented in references (9) and (10).

Figure 3 indicates the errors which may arise from conduction alone under certain typical gas-turbine operating conditions, for Nos. 22 and 14 bare Ch-Alu thermocouples and for a stainless steel thermocouple well 0.25″ o.d. x 0.17″ i.d. For all of these the conduction error is less than 2°F if the junction is immersed 2″ or more.

Loss by Radiation. In the previous section the loss of heat by

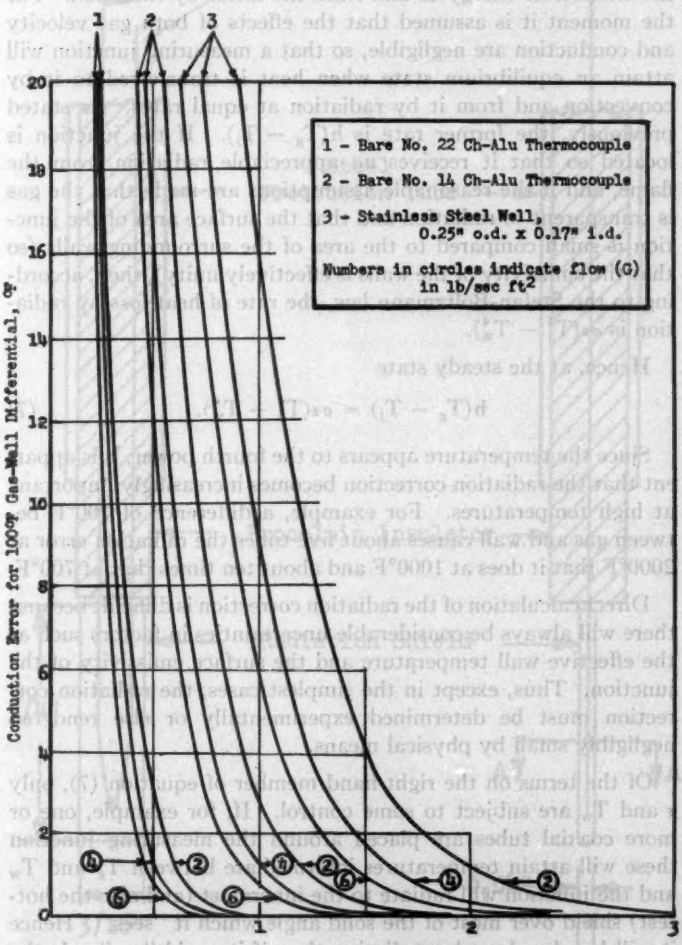


FIGURE 3.—EFFECT OF IMMERSION (A) ON CONDUCTION ERROR.

conduction from the junction was considered without regard to the transfer of energy to and from the latter by radiation. For the moment it is assumed that the effects of both gas velocity and conduction are negligible, so that a measuring junction will attain an equilibrium state when heat is transferred to it by convection and from it by radiation at equal rates. As stated previously, the former rate is $h(T_g - T_j)$. If the junction is located so that it receives no appreciable radiation from the flame, and if the reasonable assumptions are made that the gas is transparent to radiation and that the surface area of the junction is small compared to the area of the surrounding walls (so that the emissivity of the walls is effectively unity), then, according to the Stefan-Boltzmann law, the rate of heat loss by radiation is $\sigma\epsilon(T_j^4 - T_w^4)$.

Hence, at the steady state

$$h(T_g - T_j) = \sigma\epsilon(T_j^4 - T_w^4). \qquad (7)$$

Since the temperature appears to the fourth power, it is apparent that the radiation correction becomes increasingly important at high temperatures. For example, a difference of 100°F between gas and wall causes about five times the radiation error at 2000°F that it does at 1000°F and about ten times that at 700°F.

Direct calculation of the radiation correction is difficult because there will always be considerable uncertainties in factors such as the effective wall temperature and the surface emissivity of the junction. Thus, except in the simplest cases, the radiation correction must be determined experimentally or else rendered negligibly small by physical means.

Of the terms on the right-hand member of equation (7), only ϵ and T_w are subject to some control. If, for example, one or more coaxial tubes are placed around the measuring junction these will attain temperatures intermediate between T_g and T_w and the junction will radiate to the innermost (and also the hottest) shield over most of the solid angle which it "sees." Hence it will lose less heat by radiation than if it could "see" only the colder walls, and the use of coaxial radiation shields has been an effective solution to many radiation problems. The principal disadvantages of such shielding for gas turbine applications are that the shielding may be too large to go into the space available,

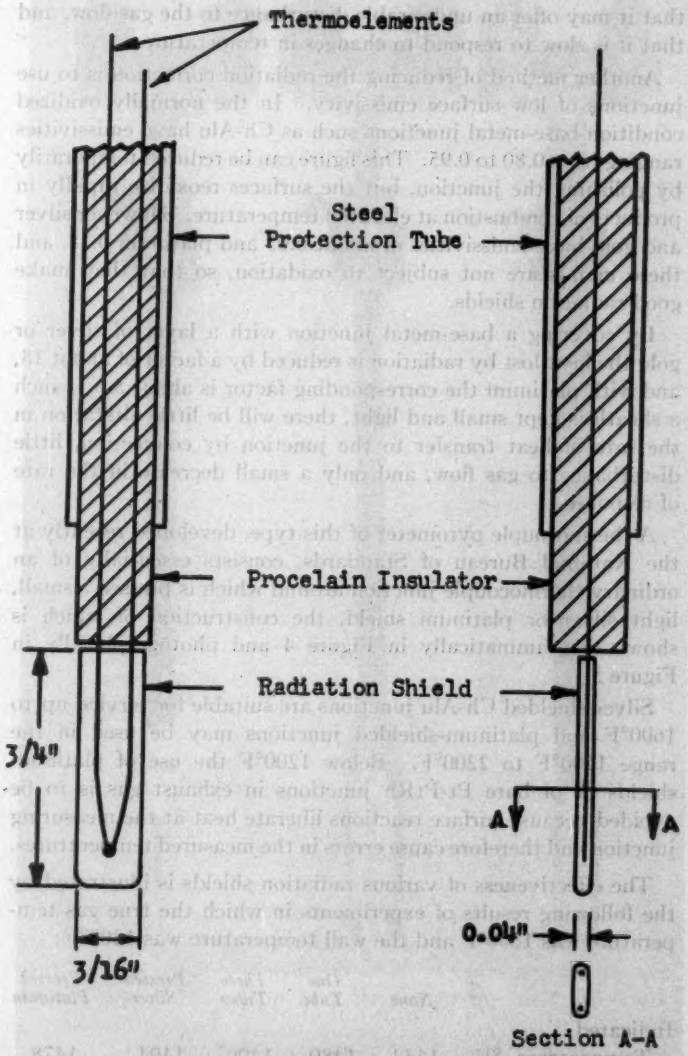


FIGURE 4.—DIAGRAM OF A JUNCTION WITH PRESSED SHIELD.

that it may offer an undesirable disturbance to the gas flow, and that it is slow to respond to changes in temperature.

Another method of reducing the radiation correction is to use junctions of low surface emissivity. In the normally oxidized condition base-metal junctions such as Ch-Alu have emissivities ranging from 0.80 to 0.95. This figure can be reduced temporarily by polishing the junction, but the surfaces reoxidize rapidly in products of combustion at elevated temperature. However silver and gold have emissivities of about 0.05 and platinum 0.18, and these metals are not subject to oxidation, so that they make good radiation shields.

By covering a base-metal junction with a layer of silver or gold the heat lost by radiation is reduced by a factor of about 18, and with platinum the corresponding factor is about 5. If such a sheath is kept small and light, there will be little alteration in the rate of heat transfer to the junction by convection, little disturbance to gas flow, and only a small decrease in the rate of response.

A thermocouple pyrometer of this type, developed recently at the National Bureau of Standards, consists essentially of an ordinary thermocouple junction around which is pressed a small, light silver or platinum shield, the construction of which is shown diagrammatically in Figure 4 and photographically in Figure 5.

Silver-shielded Ch-Alu junctions are suitable for service up to 1600°F and platinum-shielded junctions may be used in the range 1200°F to 2200°F. Below 1200°F the use of platinum shields or of bare Pt-PtRh junctions in exhaust gas is to be avoided because surface reactions liberate heat at the measuring junction and therefore cause errors in the measured temperatures.

The effectiveness of various radiation shields is illustrated by the following results of experiments in which the true gas temperature was 1500°F and the wall temperature was 1200°F:

	None	*One Tube*	*Three Tubes*	*Pressed Silver*	*Pressed Platinum*
Indicated Temperature, °F	1432	1480	1490	1494	1478
Radiation Correction, °F	68	20	10	6	22

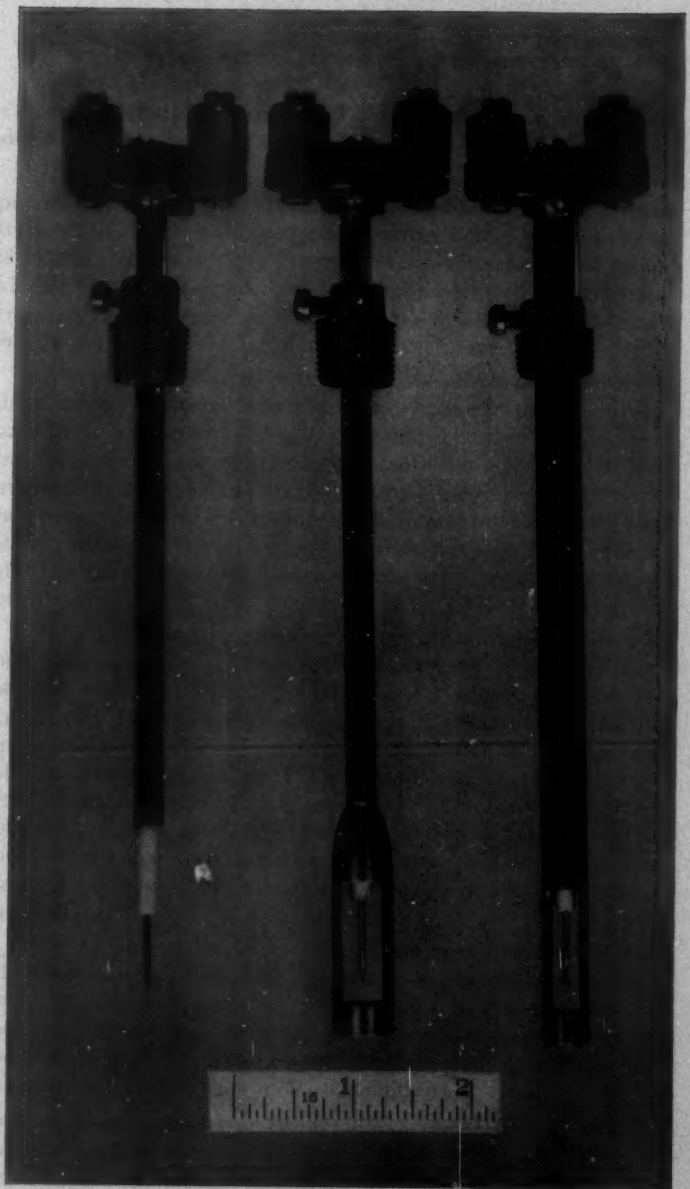
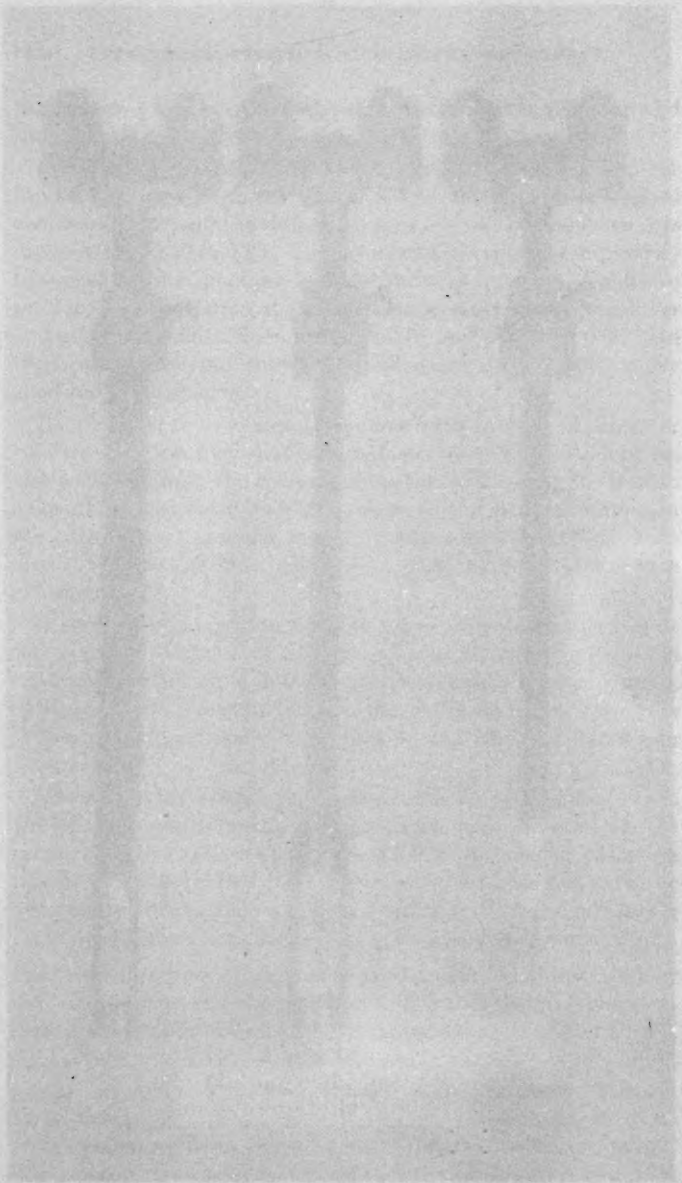


FIGURE 5.—THREE TYPICAL SHIELDED THERMOCOUPLES.



A more detailed description of the instruments with pressed shields is being prepared for publication.

Effects of Gas Velocity. According to Maxwell (11) "The temperature of a body is its thermal state considered with reference to its ability to communicate heat to other bodies." For gases it is advantageous to specify "thermal state" in terms of the mean kinetic energy of the molecules, and to consider that the temperature of a gas in thermal equilibrium is the measure of this kinetic energy.

In the previous sections of this paper the effects of gas velocity upon the temperatures indicated by a junction in the gas stream have been neglected. Actually this impact effect is of minor importance in many gas turbines because the velocities are low at those locations suited to the accurate measurement of temperature. However it is not negligible in other devices including some turbo-jet and ram-jet engines, wind tunnels, high-pressure steam systems, etc., and it may not be negligible in gas turbines of the future.

The temperature of a gas in thermal equilibrium, be it quiescent or moving, is a definite, measurable property which, by definition, is directly proportional to the mean kinetic energy of the molecules. To measure gas temperature it is common practice to immerse an instrument, such as a thermometer or the junction of a thermocouple, in the gas in such a manner that its temperature becomes approximately equal to that of the gas. The primary problem in the measurement of gas temperature is to bring the temperature-sensitive instrument to a state which is truly indicative of the total kinetic energy of the average gas molecule.

This involves proper evaluation of the heat transfer between the junction and its surroundings by conduction and by radiation, whether the gas be quiescent or in motion. In the latter case an additional factor must be considered when the uniform, directed velocity is high enough to be significant in comparison with the mean velocity of random molecular motion.

If u is the mean velocity of random molecular motion, the internal kinetic energy of unit mass of quiescent gas is $u^2/2$ and its temperature is, by definition, directly proportional to u^2. When the directed velocity is V, the kinetic energy of directed

mass motion of unit mass is $V^2/2$, and its total kinetic energy is $(u^2 + V^2)/2$.

It has been found useful to designate the quantity which is proportional to $u^2 + V^2$ as the total, dynamic or stagnation temperature, as distinguished from the ordinary or static temperature of the gas at rest. Although it can be argued that total temperature is not truly a temperature according to Maxwell's definition, this fact does not lead to errors of application, and since the expression has attained such wide usage, a change in its name would only lead to confusion.

The physical significance of static and total temperature can be understood by considering that the former would be indicated directly by a thermocouple moving in the same direction and at the same velocity as the gas, while the latter would be indicated by one having a fixed position in the stream and capable of stopping completely and adiabatically the directed motion of all of the gas which impinges upon it. Since moving the instrument with the gas is impracticable, the static temperature of a flowing gas is not measured. However, total temperature can be measured directly by a properly designed instrument.

Essentially, such an instrument consists of a temperature-sensitive element located within a cup which is open at its upstream end and closed, or nearly so, downstream. A simple type is shown diagrammatically in Figure 6, and types with additional refinements are discussed by Hottel and Kalitinsky (12).

Since the gas is stopped completely within the cup, the measuring junction attains the total temperature of the gas, provided there is no heat transfer to or from the surroundings. Instruments of this type are used frequently in wind tunnels, where there is usually plenty of space, where the temperatures are so low that the junctions do not need to be replaced frequently, and where sealing is simple because the differences in pressure are small. However, in gas turbines and steam systems where installations must be made in odd-shaped and limited spaces, where it is desirable to change elements frequently because of the high temperature involved, and where it is necessary to seal against high pressures, instruments of the cup-type have many practical disadvantages. Hence, in these latter applications it is often desirable to use bare, shielded or pencil-type junctions, or to

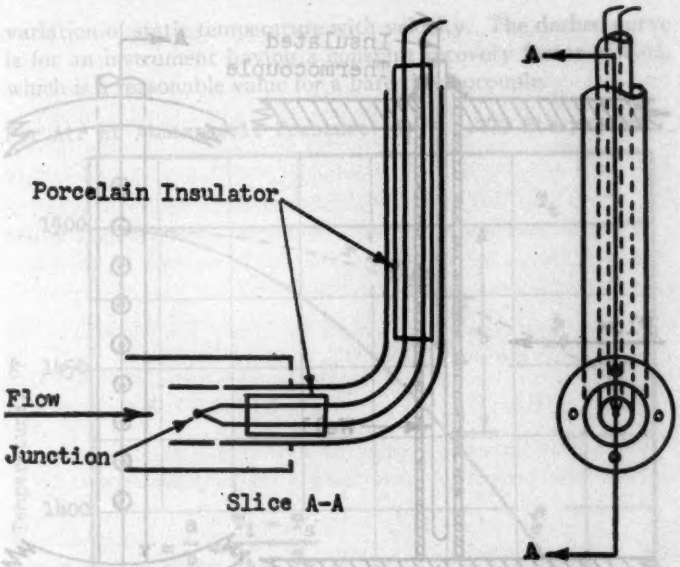


FIGURE 6.—DIAGRAM OF SIMPLE CUP-TYPE THERMOELEMENT.

install wells which completely seal the gas from the element. The latter may be a part of a modified cup-type thermoelement, as illustrated in Figure 7. Each of these, when immersed in a stream of moving gas, will attain a different temperature, even though heat interchange with the surrounding walls is eliminated. The actual value of the indicated temperature will depend upon several factors worthy of separate consideration.

If there is no transfer of heat by conduction or radiation, the temperature of a junction in a flowing gas will be determined by three processes, namely (a) the transfer of the kinetic energy of random molecular motion by collision; (b) the transfer of kinetic energy of directed motion by impact; and (c) the transfer of heat generated by friction due to the flow of gas around the instrument.

If processes (b) and (c) did not occur, the junction would attain a temperature proportional to u^2, that is the static temperature of the gas. If, in addition to process (a), all of the directed kinetic energy were transferred by impact and without friction, the junction would attain the total temperature of the gas, as is

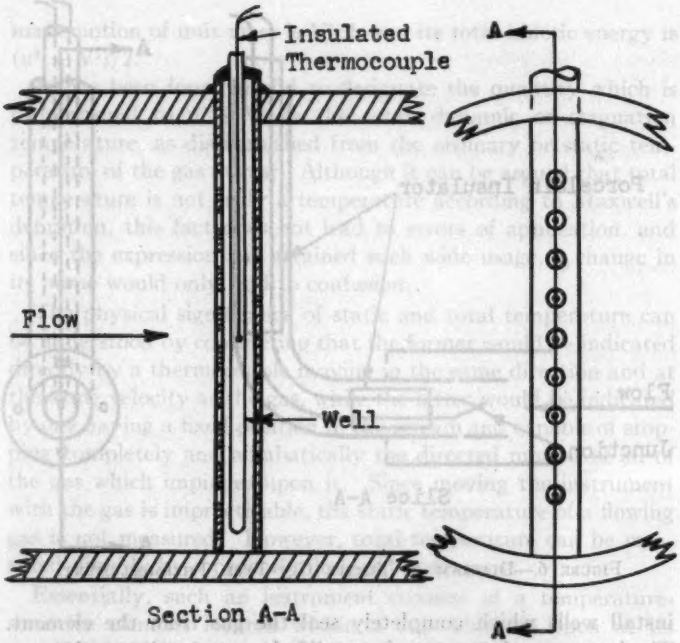


FIGURE 7.—DIAGRAM OF MODIFIED CUP-TYPE THERMOELEMENT.

the case with the cup-type junction. However many types stop the flow completely only on those areas which are normal to the direction of flow, while over the remainder of the surface the conversion is incomplete and flow continues. Because of the incomplete conversion, the junction is not heated to the total temperature. On the other hand the heat generated by friction in the boundary layer flowing past the junction tends to increase its temperature. These two effects of opposite sign would be difficult to evaluate individually, but their resultant effect, which is the item of practical importance, can be determined experimentally. It is commonly called the recovery factor of the instrument, and is defined as $r = (T_i - T_s)/(T_t - T_s)$.

The magnitude of the difference between the total and the static temperature of air with a total temperature of 1500°F is illustrated in Figure 8, which also applies approximately for the working medium of gas turbines. The bottom curve shows the

variation of static temperature with velocity. The dashed curve is for an instrument having a constant recovery factor of 0.65, which is a reasonable value for a bare thermocouple.

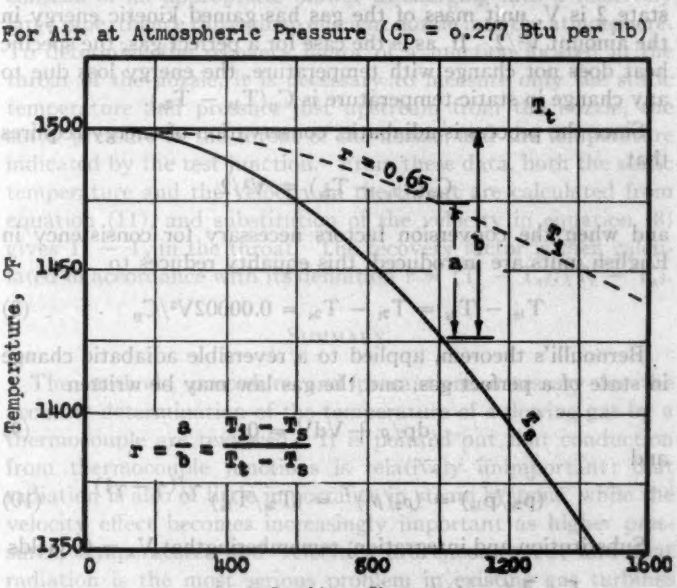


FIGURE 8.—RELATION BETWEEN TOTAL AND STATIC TEMPERATURES OF AIR AT VARIOUS VELOCITIES.

The recovery factor of a specific instrument depends primarily upon its physical configuration and does not vary much with gas velocity. Experimental values of recovery factor determined in air which has not been dried become somewhat irregular at overall velocities above 700 ft per sec. It is believed that these effects are caused by condensation of water near the instrument, producing localized thermal effects.

In order to derive the relations between static and total temperature, consider that a gas in a large reservoir at state 1, is accelerated, as by passing through an ideal converging nozzle with no friction or heat exchange with the surroundings, so that it reaches state 2. Since the reservoir is assumed to be large, the velocity of the gas at state 1 is essentially zero, its static and total

temperatures are equal and can be measured directly. Since passage through the nozzle is adiabatic, no change in total temperature can occur, so that $T_{2t} = T_{1t} = T_{1s}$. If the velocity at state 2 is V, unit mass of the gas has gained kinetic energy in the amount $V^2/2$. If, as is the case for a perfect gas, the specific heat does not change with temperature, the energy loss due to any change in static temperature is $C_p(T_{1s} - T_{2s})$.

Since the process is adiabatic, conservation of energy requires that

$$C_p(T_{1s} - T_{2s}) = V^2/2,$$

and when the conversion factors necessary for consistency in English units are introduced, this equality reduces to

$$T_{1s} - T_{2s} = T_{2t} - T_{2s} = 0.00002V^2/C_p \tag{8}$$

Bernoulli's theorem, applied to a reversible adiabatic change in state of a perfect gas, and the gas law may be written

$$dp/\rho + VdV = 0, \tag{9}$$

and

$$(p_{2s}/p_{1s}) = (\rho_2/\rho_1)^{\gamma} = (T_{2s}/T_{1s})^{\gamma/(\gamma-\gamma 1)} \tag{10}$$

Substitution and integration, rememberingthat $V_1 = 0$, yields

$$\left[\frac{p_{2s}}{p_{1s}}\right]^{(\gamma-1)/\gamma} = \frac{T_{2s}}{T_{1s}} = 1 - \frac{V_2^2(\gamma-1)}{2\gamma}\,\frac{\rho 1}{p_{1s}} \tag{11}$$

If the conditions of the gas at state 1 and its velocity at state 2 are measured, $T_{2t} - T_{2s}$ can be calculated by equation (8) and T_{2s} can be calculated by equation (11).

An actual test system can be operated under conditions which approximate very closely the ideal conditions assumed in the derivation. It can be shown that the gas velocity at state 1 need only be kept below 100 ft per sec. to avoid significant error from this source. From the experimental standpoint it is undesirable to measure V_2 directly, as with a pitot tube, because of the effect of the latter on the flow. Fortunately such a measurement is not necessary, since V can be calculated, in accordance with equation (11), from observed values of the static pressures at the two states.

Thus a practical system in which recovery factors may be measured in air under approximately atmospheric conditions, consists of an appropriate blower discharging into a relatively large pipe, thence through an efficient nozzle into the atmosphere. To determine the recovery factor of a junction located in the throat of the nozzle, it is necessary to measure only the static temperature and pressure just upstream from the nozzle, the static pressure in the throat of the nozzle, and the temperature indicated by the test junction. From these data, both the static temperature and the velocity in the throat are calculated from equation (11), and substitution of the velocity in equation (8) gives $T_t - T_s$ in the throat. The recovery factor is then calculated in accordance with its definition $r = (T_i - T_s)/(T_t - T_s)$.

SUMMARY.

The methods, procedure and precautions necessary for the accurate determination of the temperature of a flowing gas by a thermocouple are reviewed. It is pointed out that conduction from thermocouple junctions is relatively unimportant; that radiation is also of little importance in steam systems, while the velocity effect becomes increasingly important as higher pressures, temperatures and velocities are encountered; and that radiation is the most serious problem in existing gas turbines and jet engines. The independent effects of such factors as physical size and shape, temperature difference between gas and wall, depth of immersion, gas velocity, etc., are discussed by reference to equations and charts.

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FOUNDRY CONTROL TEST FOR MELTING HIGH ZINC BRONZES AND BRASSES (MANGANESE BRONZE, NAVAL BRASS, COMMERCIAL BRASS).

By Joseph A. Duma[1].

In most brass foundries, the practice of making furnace additions of zinc to compensate for the zinc volatilized out of the melt by too long flaring, or too high a furnace temperature, or both, are at best based on empirical formulas which sometimes hit, but more often than not, miss the object which they aim to attain. For example, the foundry with which the author is associated, for a long time had employed the practice of adding to all heats prior to tapping 1½ lb. zinc per 100 lb. of metal (manganese bronze) charged. This practice periodically caused approximately 40 per cent of the heats to fail with respect to physical properties. Either the tensile strength would be too high and the elongation too low, or vice versa. Needless to add, to permit too many runs of such high rejects is to invite economic suicide.

A search was authorized to find some one physical property which changed critically and yet appreciably with accompanying small changes in chemical composition. Several physical properties which were suspected to behave in this manner were investigated for control purposes. Correlation between chemistry and electrical resistance measurements, melting points, the position of the liquidus and solidus points and phase changes detected pyrometrically yielded no satisfactory procedure of control, for the tests involved were either too cumbersome, impractical, or too time consuming. The chemical laboratory with all its facilities which, among other speed devices, included a modern type spectrograph and a pneumatic tube system for transportation of specimens and results, could not inform us regarding the composition of the melt within the short space of time which elapses between the melting and pouring of bronzes.

Before abandoning the project, it was decided to try to correlate specific gravity with tensile strength and elongation for a

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control test. Most unexpectedly, this turned out to be the fastest as well as the most effective control test yet found. Since its implementation into standard practice, not a single heat of manganese bronze, naval brass, or commercial brass has failed with respect to physical properties. Several hundred heats of metal have been made using the specific gravity method of control. The test has freed us from considerable anxiety and is saving us a great deal of money in that no castings are rejected for failure to pass specified minimum physical properties.

To make this test the following comparatively inexpensive equipment is needed:

A laboratory balance sensitive to 2 mg.

A set of gram weights.

A copper specimen mold (See Figures 1 and 2).

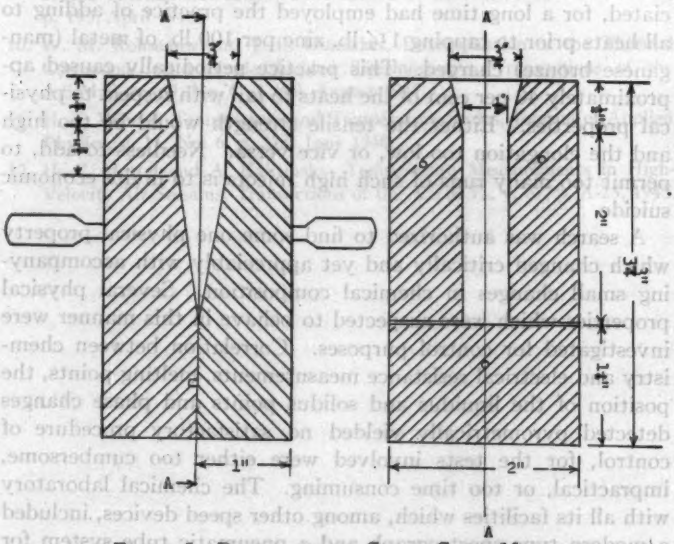


FIGURE 2.—DIMENSIONS OF COPPER CHILL MOLD.

In practice, the manner in which the test is performed is as follows. As soon as all the metal constituting the charge is melted, the furnace operator or melter ladles out of the furnace a hand-ladleful of metal which he pours into the copper test mold. Immediately after pouring it, he parts the mold in two,

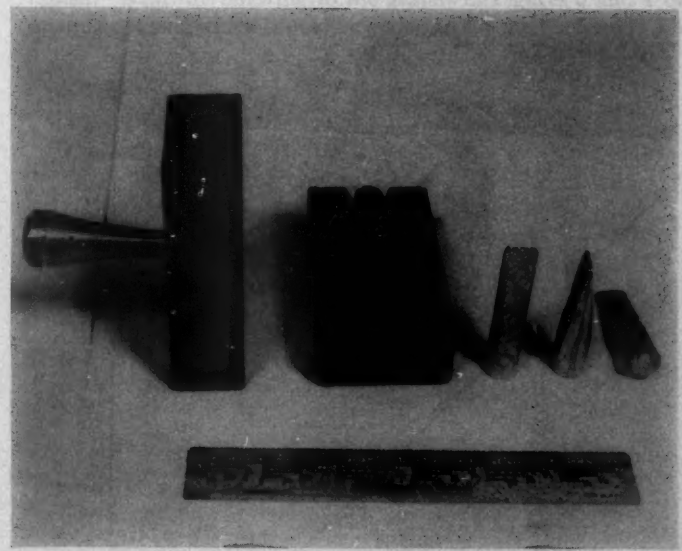
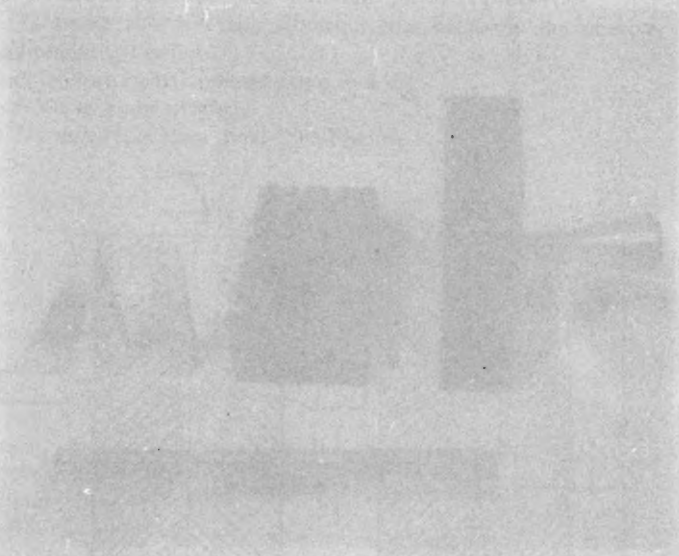


FIGURE 1.—COPPER CHILL MOLD AND SPECIMENS MADE THEREIN.



knocks the specimen out, and takes it to a nearby laboratory. A laboratory workman with a hand-hacksaw then removes the top half inch discard, trims off any fins attached to the specimen on the belt surfacer, and finally weighs the specimen both in air and in water (preferably distilled). By simple calculation, which involves subtraction (weight in air minus weight in water) and division (loss in weight as obtained above divided into the weight of the specimen in air), the specific gravity of the metal is obtained. Next, by referring to the graphs in Figures 3 and 4 the quantity of zinc which should be added to the melt is found. The

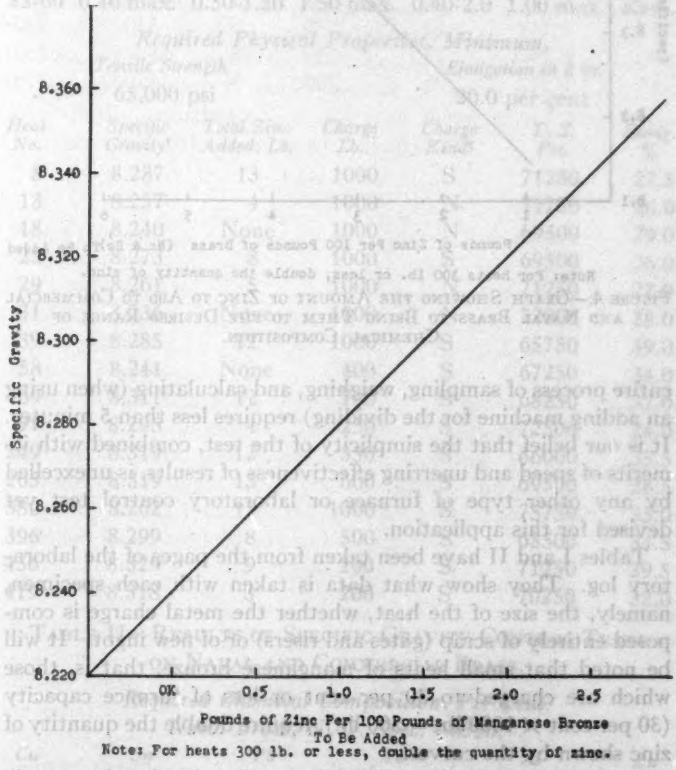


FIGURE 3.—GRAPH SHOWING THE AMOUNT OF ZINC TO ADD TO MANGANESE BRONZE TO BRING IT TO THE DESIRED RANGE OF CHEMICAL COMPOSITION.

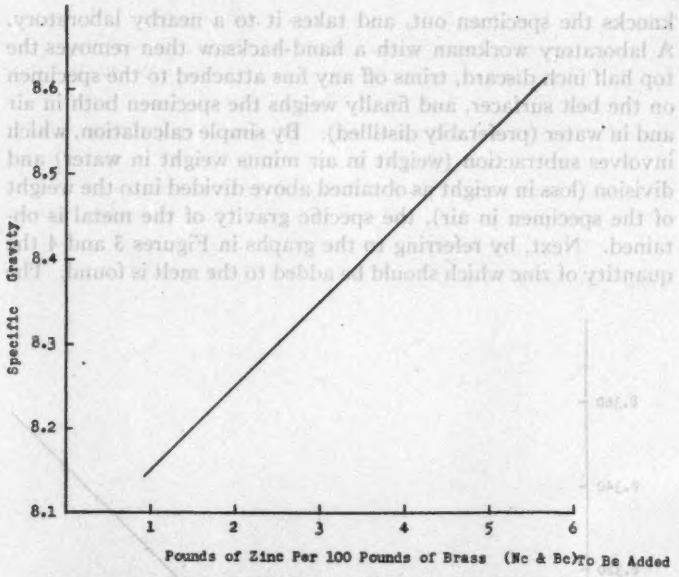


FIGURE 4.—GRAPH SHOWING THE AMOUNT OF ZINC TO ADD TO COMMERCIAL AND NAVAL BRASS TO BRING THEM TO THE DESIRED RANGE OF CHEMICAL COMPOSITION.

entire process of sampling, weighing, and calculating (when using an adding machine for the dividing) requires less than 5 minutes. It is our belief that the simplicity of the test, combined with its merits of speed and unerring effectiveness of results, is unexcelled by any other type of furnace or laboratory control test yet devised for this application.

Tables I and II have been taken from the pages of the laboratory log. They show what data is taken with each specimen, namely, the size of the heat, whether the metal charge is composed entirely of scrap (gates and risers) or of new ingot. It will be noted that small heats of manganese bronze, that is, those which are charged to 30 per cent or less of furnace capacity (30 per cent × 1000 lb. = 300 lb.), require double the quantity of zinc shown by the curves.

In conclusion, this control test is considered to be of economic advantage to all brass foundries making navy type brasses under

Navy Department Specifications (Mn-c, N. D. Spec. 49B3; N-c, N. D. Spec. 46B10; B-c, N. D. Spec. 46B11) to avail themselves of this simple, practical, and inexpensive control test to insure passing the physical property requirements of said specifications.

TABLE I—RESULTS OF SPECIFIC GRAVITY CONTROL TESTS ON MANGANESE BRONZE.

Manganese Bronze, Navy Dept. Spec. 49B3.

Required Chemical Composition, Per Cent.

Copper	*Lead*	*Aluminum*	*Manganese*	*Iron*	*Tin*	*Zinc*
55-60	0.40 max.	0.50-1.50	1.50 max.	0.40-2.0	1.00 max.	Rem.

Required Physical Properties, Minimum.

Tensile Strength	*Elongation in 2 in.*
65,000 psi	20.0 per cent

Heat No.	*Specific Gravity*[1]	*Total Zinc Added, Lb.*	*Charge Lb.*	*Charge Kind*[2]	*T. S. Psi.*	*Elong. %*
8	8.287	13	1000	S	71250	27.5
13	8.257	4	1000	N	71250	26.0
18	8.240	None	1000	N	69500	29.0
25	8.273	8	1000	S	69500	26.0
29	8.261	5	1000	N	71250	27.0
31	8.236	None	1000	N	72000	28.0
39	8.285	12	1000	S	65750	39.0
58	8.241	None	400	S	67250	34.0
69	8.311	12	700	S	68250	32.0
91	8.295	4	250	S	72500	24.0
249	8.319	12	350	S	66000	35.5
265	8.319	13	600	S	69500	28.5
386	8.262	17	1000	S	75750	25.0
396	8.299	8	500	S	68500	34.5
456	8.329	9	200	S	71250	29.5
479	8.318	8	200	S	70250	32.0

TABLE II—RESULTS OF SPECIFIC GRAVITY CONTROL TESTS ON NAVAL AND COMMERCIAL BRASS.

Required Chemical Composition, Per Cent.

Naval Brass, N. D. Spec. 46B10.

Cu	*Sn*	*Pb*	*Fe*	*Al*	*Zn*
60-65	0.50-1.50	0.75-1.50	0.75 max.	0.50 max.	Rem.

[1] Not final specific gravity, but that immediately after melting.
[2] N signifies new, ingot metal. S signifies scrap metal.

Commercial Brass, N. D. Spec. 46B11, Gr. I.

65-70	1.50 max.	1.50-3.0	0.75 max.	0.30 max.	Rem.

Required Physical Properties, Minimum.

	T. S.	*Elong. in 2 in.*
Naval Brass	30,000 psi	15.0 per cent
Commercial Brass	None	None

Heat No.	*Specific Gravity*	*Total Zn Added, Lb.*	*Charge Lb.*	*Charge Kind*	*T. S. Psi.*	*Elong. %*
86	8.512	10	150	S	43000	40.5
171	8.577	20	250	S	48000	40.5
277	8.414	21	550	S	55000	36.0
289	8.497	24	350	S	52750	28.0
381	8.342	21	700	N	45000	27.0

SOME ADVANTAGES OF PLANETARY GEARS.

By W. G. Stoeckicht.*

Planetary gears, in order to work satisfactorily, need special means of securing automatically an equal load distribution between the planets. Unless such means are provided, even the highest obtainable accuracy in producing the gears will not be able to guarantee this equal load distribution.

This is made clear by the fact that, in a planetary gear having 3 planets, at least 17 major sources of error, which cumulate periodically, contribute to a poor load distribution. The following table will indicate these sources of error.

Table No. 1.
ERRORS.

Part of Gearing	*Spacing or Pitch*	*Shape of Involute*	*Eccentricity*	*Sum of Errors*
Sun Gear	1	1	1	3
3 Planets	3	3	3	9
Internal Gear	1	1	1	3
Planet Carrier	1		1	2
Sum of Errors	6	5	6	17

Errors resulting from these sources can be reduced to pitch errors or errors in action. Table No. 2 shows how the different sources of error contribute to the resulting error in action, considering a single tooth contact within the planetary gear. (The factor X2 indicates that the amount of error will appear doubled when reduced to error in action.

Table No. 2.
ERRORS.

Part of Gearing	*Spacing or Pitch*	*Shape of Involute*	*Eccentricity*	*Sum of Errors*
Sun Gear	1	1	1	3
Planet Gear	1 X 2	1 X 2	1 X 2	6
Internal Gear	1	1	1	3
Planet Carrier	1 X 2		1 X 2	4
Sum of Errors	6	4	6	16

* The author has been awarded a Diploma of Honour (Diplome D' Honneur) on the Paris World Exhibition 1937 for the planetary gear of the 1400 Hp Diesel Locomotive exhibited by the German Railways.

Table No. 3 indicates the amount of the possible error in action, if the single errors do not exceed the following values:

Spacing or pitch error of gears	.0002″
Error in shape of involute	.0002″
Eccentricity of gears	.0005″
Pitch error of planet carrier	.0005″
Eccentricity of planet carrier	.0005″

These dimensions of errors can be considered as the very lowest at the present time obtainable in production jobs.

TABLE NO. 3.

All dimensions in thousandths of an inch.
Dimensions of error reduced to pitch errors.

Part of Gearing	*Spacing or Pitch*	*Shape of Involute*	*Eccentricity*	*Sum of Errors*
Sun Gear	.2	.2	.5	.9
Planet Gear	.4	.4	1.0	1.8
Internal Gear	.2	.2	.5	.9
Planet Carrier	1.0	..	1.0	2.0
Sum of Errors	1.8	.8	3.0	5.6 = maximum error in action

It is shown that, although considering the highest accuracies obtainable, an error in action of a very large dimension will appear. This error in action may vary periodically between zero and a maximum value, and it will prevent equal load distribution unless special means are provided.

An unequal load distribution means that, at least periodically, the whole load may have to be carried by one single planet. Therefore, in order to avoid failure, the gearing would have to be dimensioned accordingly, and no advantage in dimensions and weight would be gained, compared with the usual type of non-planetary gearing.

This survey deals with planetary gears having special means for providing automatically an equal load distribution on the planets, such as have been developed by the author. They have proved to be very successful and, compared with the usual type of non-planetary gears, they show many advantageous features, the more important of which will be discussed in the following paragraphs.

Dimensions and Weight.

In a planetary gear the total load is divided between several planets and this is the main reason for the small dimensions of planetary gears. If, for instance, 3 planets are used, each tooth engagement on the sun gear will have to carry one-third of the whole load. That means, roughly speaking and not considering some influences of secondary importance, that the sun gear will need to have the dimensions of a pinion designed for about one-third of the actual load.

This sun gear will have a smaller circumferential speed in proportion to its reduced diameter. However, in nearly all instances of application, the planet carrier turns in the same direction as the sun gear, and therefore the actual relative tooth speed results from the difference of RPM of sun gear and planet carrier. Thus, the tooth engagement speed for the planetary gear will always be considerably lower than that of a comparable plain (non-planetary) gear. As additional dynamic tooth loads increase with the tooth engagement speed, the planetary gear will have to provide less additional strength with regard to such dynamic loads. It also seems to be important to call attention to the fact that the equal load distribution means that errors in action at the different points of tooth engagement are reduced to an average error in action, which in turn results in causing smaller additional dynamic tooth loads.

It is a commonly accepted fact that the use of hardened gears in high precision work is limited to comparatively small gear dimensions because of distortions due to heat treatment and because there is very little machinery available for grinding large gears. The small dimensions of the single pinions (sun and planets) will allow them to be provided as hardened gears thus making advantageous use of their high load carrying qualities.

All these factors combine to facilitate the application of small dimension gears to large horsepower transmission. Of course, no particular ability is required for reducing gear dimensions simply by increasing specific stresses. But, as shown above, properly designed planetary gears will enable us to build gears of very small dimensions without increasing specific stresses beyond the level of what can be considered safe and reliable.

Apart from what has been mentioned above, the concentric arrangement of the planetary set provides a very compact unit. Also, in gears with a driven or driving planet carrier, the reduction ratio is greater than the figure given by the quotient of the number of teeth on the inner and outer sun wheels. (If N_g and N_p are the number of teeth of sun gear (driving member) and ring gear or outer sun wheel (stationary member) respectively, the gear ratio R is defined by the equation $R = \frac{N_g}{N_p} + 1$).

Dimensions and Weight.

Because of the small dimensions obtainable with properly designed planetary gears, the weight of such units and the amount of steel required for their production are low. Furthermore, the correct mesh of the gear teeth does not depend on the stiffness or rigidity of the gear casing. This casing, apart from transmitting the reaction torque of the gear, does not get any stresses from the single tooth loadings, with the exception of thrust, when single helical gears are used. Therefore light weight casings can be provided for planetary gears without endangering their operating characteristics.

In order to show the progress that has been achieved through the application of planetary gears, the design data of two gears, used for marine propulsion, are outlined as follows:

1. A turbine reduction gear.
   Maximum load: 5000 Shp.
   Reduction from 3770 to 580 Rpm.
   Weight: 2200 lbs.
   Outer diameter of casing: 35⅜ inches.
   Length between coupling flanges: 29½ inches.
   A comparable non-planetary gear, designed as a double reduction gear in order to obtain small dimensions, still would have a weight of approximately 10,500 lbs., or 4.77 times as much as the planetary gear.
2. A reduction gear to be flanged to a marine diesel engine.
   Maximum (continuous) load: 2500 Hp.
   Reduction from 670 to 470 Rpm.
   Weight: Approximately 1300 lbs.
   Outer diameter of casing: 26.4 inches.
   Length between coupling flanges: 25.5 inches.

FIGURE 1.—SHOWS A VIEW OF THE ASSEMBLED GEAR.

FIGURE 2.—SHOWS THE VERY SIMPLE GEAR CASING.

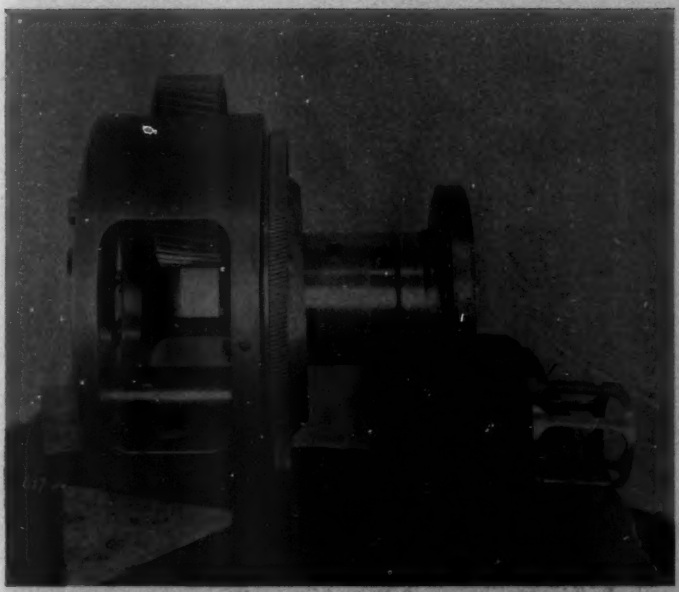


FIGURE 3.—SHOWS THE PLANET CARRIER WITH ONE PLANET PINION AND A FURTHER PLANET PIN.

FIGURE 4.—SHOWS THE SUN GEAR, ONE PLANET AND THE INTERNAL GEAR RING INCLUDING THE FLEXIBLE ATTACHMENT FOR THE LATTER.

A comparable non-planetary gear would weigh approximately 9000 lbs. or 6.9 times as much as the planetary gear.

Sectional drawings of the gear referred to under No. 1 are shown on Plates 1 and 2. From Plate 1 it will be seen that a driving sun gear (1) is in mesh with 3 planet pinions (2), which in turn mesh with an internal ring gear (3). This ring gear (3) is coupled to the casing by means of a double tooth coupling (5, 6, 7 on Plate 2), thus enabling it to float freely. This freedom of self-adjustment provides a perfect balance between the tooth loads. The photographs should clarify gear construction.

Fig. 1. Shows a view of the assembled gear.

Fig. 2. Shows the very simple gear casing.

Fig. 3. Shows the planet carrier with one planet pinion and a further planet pin.

Fig. 4. Shows the sun gear, one planet and the internal gear ring including the flexible attachment for the latter.

Running Conditions and Performance.

As mentioned above, a planetary gear designed for the transmission of a fixed horsepower for a given speed of driver compared with a plain gear (non-planetary) consisting of pinion and bull gear, will have smaller pinion diameter and smaller tooth contact speed. It is true today that very high tooth speeds have proved to work satisfactorily; nevertheless, if lower tooth speeds can do the same work, this may be regarded as an advantageous feature.

In the two instances given above the tooth speeds compare as follows:

(1) 5000 Hp. Planetary gear.....4430 Ft/min.
Non-planetary comparable gear.....8880 Ft/min.

(2) 2500 Hp. Planetary gear.....1040 Ft/min.
Non-planetary comparable gear.....4390 Ft/min.

In a planetary gear with an equal load distribution between the planets the tooth loadings on each sun gear are held in equilibrium; therefore their respective bearings will have to carry no load (except thrust resulting from tooth helix when single helical gears are used).

In summarizing, several features contribute to the good performance of the planetary gear:

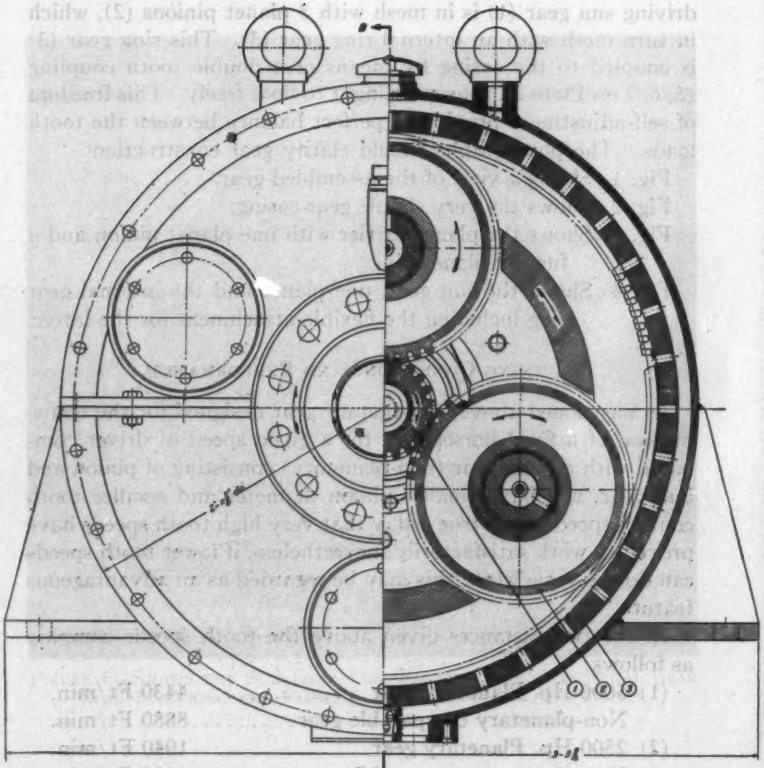


PLATE 1

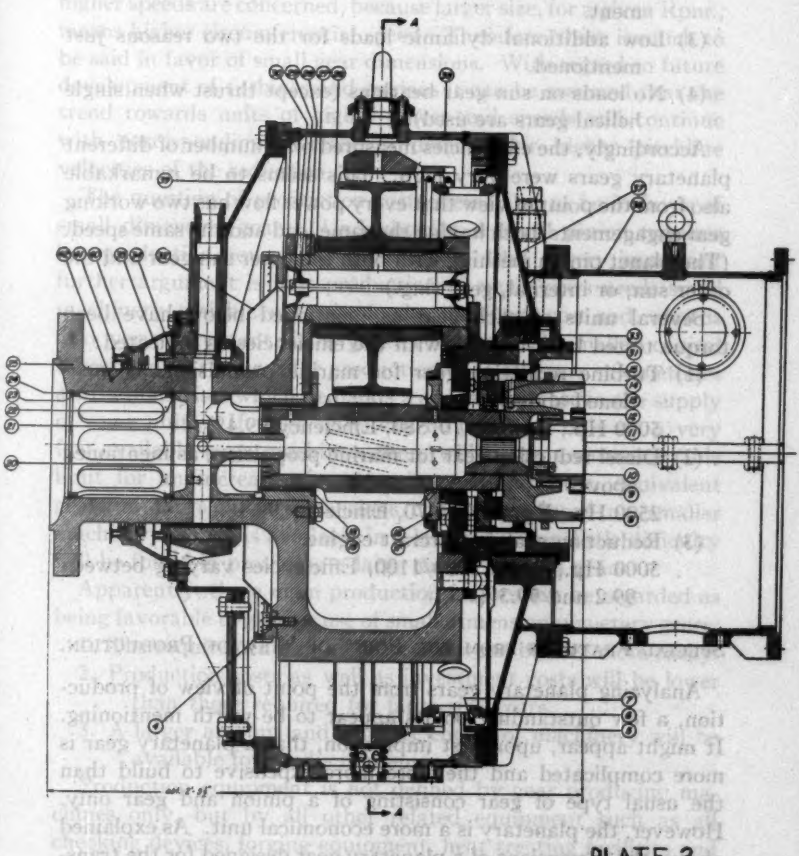


PLATE 2

(1) Tooth engagement speeds are low.
(2) Equal load distribution between the planets by self-adjusting means results from an equalization of the errors in action at the different points of tooth engagement.
(3) Low additional dynamic loads for the two reasons just mentioned.
(4) No loads on sun gear bearings (except thrust when single helical gears are used).

Accordingly, the efficiencies measured on a number of different planetary gears were very high. This seems to be remarkable also from the point of view that every power flow has two working gear engagements, both having the same load and the same speed. (The planet pinion meshing with both the inner sun gear and the outer sun, or internal, gear ring.)

Several units of each gear as mentioned below have been torque tested back-to-back with the efficiencies as indicated:

(1) Turbine reduction gear for marine propulsion, as mentioned above.
5000 Hp., Rpm. 3770/580, Efficiency 99.4%.
(2) Diesel reduction gear for marine propulsion, as mentioned above.
2500 Hp., Rpm. 670/470, Efficiency 99.4%.
(3) Reduction gear for aircraft engine.
3000 Hp., Rpm. 3000/1100, Efficiencies varying between 99.2 and 99.3%.

Special Features from the Point of View of Production.

Analyzing planetary gears from the point of view of production, a few outstanding points appear to be worth mentioning. It might appear, upon first impression, that a planetary gear is more complicated and therefore more expensive to build than the usual type of gear consisting of a pinion and gear only. However, the planetary is a more economical unit. As explained above, the dimensions of a planetary gear designed for the transmission of a given horsepower will always be very much smaller than those of the comparable non-planetary gear. Gear production is a matter of accuracy. Undoubtedly, it is less difficult to achieve high accuracy on small gears than on large ones. Spacing or pitch errors have to be measured in inches, but the indexing

accuracy of a gear production machine is an angular one. Therefore, accuracy measured on the pitch circle necessarily deteriorates with increasing diameters. Unfortunately, the demand for accuracy increases with the size of the gear, as far as gears for higher speeds are concerned, because larger size, for a given Rpm., means higher circumferential speed. Therefore, there is much to be said in favor of small gear dimensions. With regard to future development of turbines and engines it can be assumed that the trend towards units of higher power and speeds will continue with corresponding increase in demand for higher pitchline velocities of the reduction gear.

The question of production costs seems also to be in favor of small dimension gears. Undoubtedly, a small pinion requires less production hours than a large one. But a very important further argument is that production of small gears needs small machinery, which can be obtained at far lower cost and, what is also very important, in a greater variety. A smaller capital investment is required for production of small gears. Sometimes a situation arises which demands a sudden increase in the supply of some units. If these units can be produced only on a very few available large machines, there will be an unsurmountable limit for an increase of production. If, however, equivalent gears of far smaller dimensions can be produced on smaller machinery, which is available in a larger number, little difficulty will be found in providing a large supply of such gears.

Apparently, three main production factors can be regarded as being favorable to greater use of small dimension planetary gears:

1. The requirements of high accuracy can be met more easily.
2. Production costs as well as investment costs will be lower than those required for large size gears.
3. A larger amount and a larger choice of machinery will be available for production equipment.

Production equipment is not defined by gear producing machines only, but by all other related equipment such as all checking devices, forging equipment, heat treating facilities and all other machinery.

Applications.

The planetary gears mentioned above have been developed for marine propulsion. In this application the saving of weight

and space is an item of major importance, and it is due not only to the small weight and dimensions of the gears themselves, but also to their high efficiency. As losses vary between 30 and 40 per cent of those of the hitherto used type of gearing, the amount of lubricating and cooling oil, the size of the oil tanks and the cooling apparatus will be reduced in the same proportion. Indirect savings of weight also can be obtained by making use of the compactness of planetary gears. As the author pointed out in a survey written more than two years ago, such gears eventually built as a unit with the thrust block can be located far aft in a ship. The shaft connecting engine and gear, running at lower torque than the propeller shaft, will correspondingly be lighter and less expensive which will also apply to its bearings.

Many auxiliary drives in ships may benefit from smaller gear dimensions. Planetary reverse gears developed by the author according to the principles explained in this article have been very successful and have been supplied in large numbers.

Also in stationary power plants gear dimensions become important. With increasing speed turbines become smaller while the gear reduction ratio becomes greater. In order to avoid a striking disproportion between the size of a turbine and its gear, smaller gears will have to be provided than those which have hitherto been used. Besides, turbine manufacturers, who are eager to save any possible loss of heat, will equally appreciate any reduction of losses in the mechanical transmission.

In many other fields planetary gears have found wide and successful application. In more than 15 years of practical experience the author has designed and delivered planetary gears in great numbers for all types of stationary installations, such as steam and hydraulic power plants and all kinds of industrial drives. They have been used in small and large diesel locomotives, as change-speed gears for railway motor coaches, and for aircraft engines.

In many applications a definite and increasing demand for lighter and smaller gears seems to appear. A properly designed planetary gear, embodying the feature of automatic load distribution, will have the best chances of satisfying this modern trend of development.

THE NAVY IN RESEARCH.

By William H. Leahy, Captain, U. S. N.*

This is a statement of the process of scientific research and the Navy's role in it. In these troubled times, it is essential that the value of scientific research be fully appreciated, so that the future of the Navy and nation may be safeguarded.

The senior historian of the Navy recently stated that "our rapid advance in science and its application to warfare" constituted one of the five major factors that carried the Navy to complete victory in World War II. Even the enemy was forced to concede this fact. On December 14, 1943, Grand Admiral Doenitz wrote: "For some months past the enemy has rendered the U-boat ineffective. He has achieved this object, not through superior tactics or strategy, but through his *superiority in the field of science.*"

This superiority in the field of science did not come overnight. Nor will it come overnight if our Fleet is challenged again. It took years of painstaking scientific research to provide the basic knowledge that the engineers called upon in World War II to give us radar, the proximity fuze, and the atomic bomb. It is taking steady painstaking research now to give us our insurance for the nation's security in the future. Providing this reservoir of basic scientific knowledge for future emergencies is the business of the Office of Naval Research.

The smoke was scarcely clearing over atom-bombed Hiroshima when the U. S. Navy began putting into effect the most extensive peacetime scientific research program ever undertaken. Almost three months before the atom bomb actually fell on Hiroshima, Secretary-of-the-Navy Forrestal had established the Office of Research and Inventions to assure the Navy a well-coordinated research effort in every field of basic science. "Wars are fought," he said, "primarily with weapons which are developed before the fighting begins. Experience demonstrates that a nation at war usually does not have time to push through a new line of funda-

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mental research and then apply that research before the outcome of the war is decided. Research requires time. Application of its result requires additional time."

Soon after its establishment, the Office of Research and Inventions got under way to gain this precious time through scientific research. Programs were planned in the following fields of basic science: nuclear physics, physics, chemistry, electronics, mechanics and materials, geophysics, fluid mechanics, mathematics, and medical sciences. Also in the naval sciences: undersea warfare, amphibious warfare, power, armament, and air warfare. The ambitious programs which were then just plans are now under way and today we are in a position to report some important accomplishments and significant progress in these fields.

In August 1946, the office was given statutory permanence by Congress as the Office of Naval Research. Today, virtually every outstanding scientific laboratory in the country, whether part of a big industrial plant or part of a university's science department, is working on some phase of the research program of ONR. And virtually every outstanding scientist in the country is contributing to this tremendous scientific effort of the Navy.

All has not been rosy, however, in our two-year history. We have had our headaches. We have had pioneering to do and we still have pioneering to do. People are often impatient. They want results quickly. They understand better the specific things they can see and feel and touch. In other words, they can be sold more easily on spending a million dollars to build another atomic bomb just a little bit more destructive than the one we already have than they can understand the reasons for spending a million dollars to build cyclotrons and synchrotrons to bombard the atom and probe the fundamental laws of the nucleus about which we know so little. We therefore must constantly sell what we are doing and remind Congress, the general public, and the Navy that the atomic bomb, for instance, is only the beginning of man's ability to harness the forces of the atom. The more tremendous forces of the nucleus still remain untapped. And only by adding to our basic scientific knowledge can we learn how to tap this energy.

I am therefore going to digress a bit to take a brief look into how we gain this scientific knowledge. Scientific knowledge can be gained only by the systematic investigation of natural phe-

nomena to determine accurately their special characteristics, the laws of nature that govern their behavior, and their relationships to other phenomena of which something is already known. Such systematic investigation is what we call basic research. It is completely theoretical; that is to say, it is not concerned with putting to practical use the knowledge it produces, but is aimed rather at rounding out what is known about any given field. As has been aptly put, "our enemy is ignorance."

Paradoxically, it has been found that this untrammeled pursuit of knowledge for its own sake leads in the long run to greater progress in the improvement of material than does development, which is applied research deliberately directed toward a given practical goal. This is true primarily because basic research is an inquiry into the unknown, the results of which cannot be foreseen, and from time to time the results that are obtained initiate radically new types of equipment. As the great British physicist and Nobel prize winner, Sir Joseph J. Thomson, put it, "applied research leads to improvements but basic research produces revolutions in materiel." Witness, for example, the way in which Clerk-Maxwell's theory of electro-magnetism led to the development of radio and radar, and how Becquerel's discovery that uranium emits radiations contributed to the development of the atomic bomb. Basic research, by adding to our knowledge of nature and of man, extends the Navy's scope of action by making possible the development of new equipment of hitherto unobtainable performance characteristics.

This is not to say that development—which is engineering in the finest sense of the term—does not make use of scientific knowledge, for it does so to an almost unlimited extent. It means, rather, that in these days of increasing competition for national power, in which each of the great competitors is drawing to the fullest extent possible on the scientific talent to which it has access, basic research has become an essential tool to be used in building national security. Basic research, and basic research alone, produces new scientific knowledge, and it is from this peripheral area that come the revolutions whereby new Naval materiel, radically superior to anything hitherto known, is made possible.

Our first point, accordingly, is that today basic scientific research is indispensable to the building and maintenance of naval strength.

This being so, it is necessary that we examine more closely the part played by basic research in the process by which new Naval materiel is produced. The complete process, from the making of a new scientific discovery to the production of a new gadget for use by the Fleet, begins with observation and proceeds by logically sequential steps to the final tests and the adoption of the new equipment for use. Observation is either the recognition of a phenomenon hitherto unnoticed or unknown or the formulation of a new hypothesis by which already known phenomena are brought into new relations to each other. As an example, Hoyt Taylor's observation at the Naval Research Laboratory that radio waves were reflected by ships passing on the Potomac can be cited; it was one of the major starting-points of the work that ultimately produced radar.

Basic research then follows observation. It represents the accumulation of new knowledge about a phenomenon by the systematic investigation of all of its characteristics and implications. It provides the scientific data without which development cannot go very far. An illustration of basic research that quickly comes to mind is the investigation now being conducted of the properties of titanium before this element can be used by the Fleet. We know that this element has weight approximately one-half that of steel and has excellent corrosion resistance against numerous environments, particularly salt water. Since it has a higher strength-weight ratio than aluminum, numerous applications for devices aboard ship and for aircraft should be forthcoming in the future. We recognize it today as one of the most promising new elements. Specifically, titanium promises to be the solution to one of the many problems involved in supersonic flight. As you know, the skin-temperatures of objects get extremely high at supersonic speeds; hence, the need for a fuselage material with a very high melting point and good corrosion resistance. Titanium has these qualities and yet we need to know more about its properties in the pure state so we can find methods of obtaining it in greater quantities. We need to know its alloying possibilities in order to endow it with superior properties. And after we have accumulated all this basic knowledge about the element, we will have to develop methods of manufacturing it on a large scale if it is to be economically useful.

In other words, after the basic research is completed, our next

FIGURE 1.—Radar antenna outlined against the sky on an *Essex*-class flattop. Radar owed its rapid development to the advent of the last war, but it was based on fundamental scientific knowledge accumulated many years before. The program of the Office of Naval Research is an attempt to replenish this stock of basic knowledge from which developments may come later in case of a national emergency.

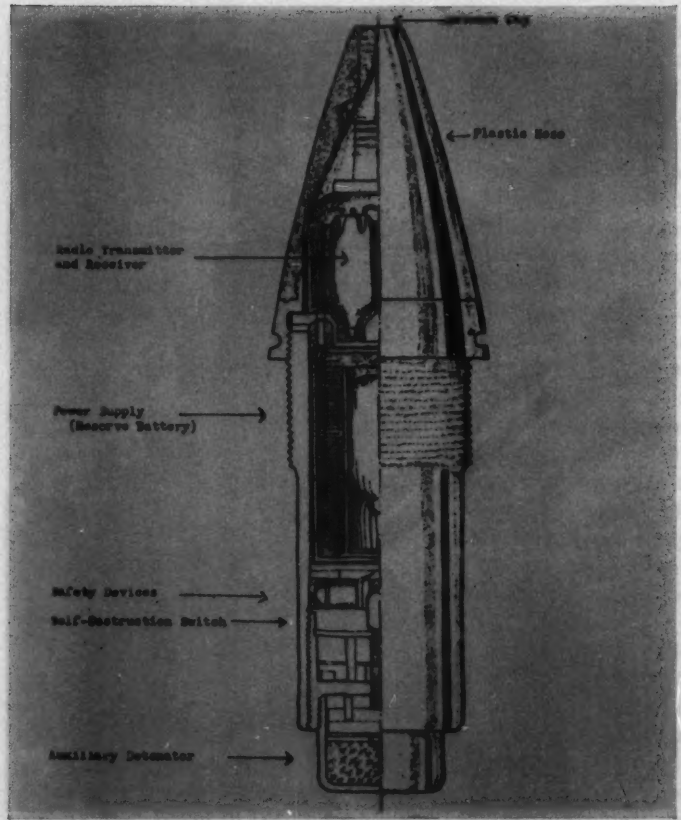


FIGURE 2.—Diagram of the proximity fuze, "an engineering miracle of the last war", whose development involved every step from basic research to test and evaluation and final production. Its development took four years and involved what amounts to development of a complete radio set, equipped with a power plant, transmitter, and receiver, all fitted into the nose of a shell.

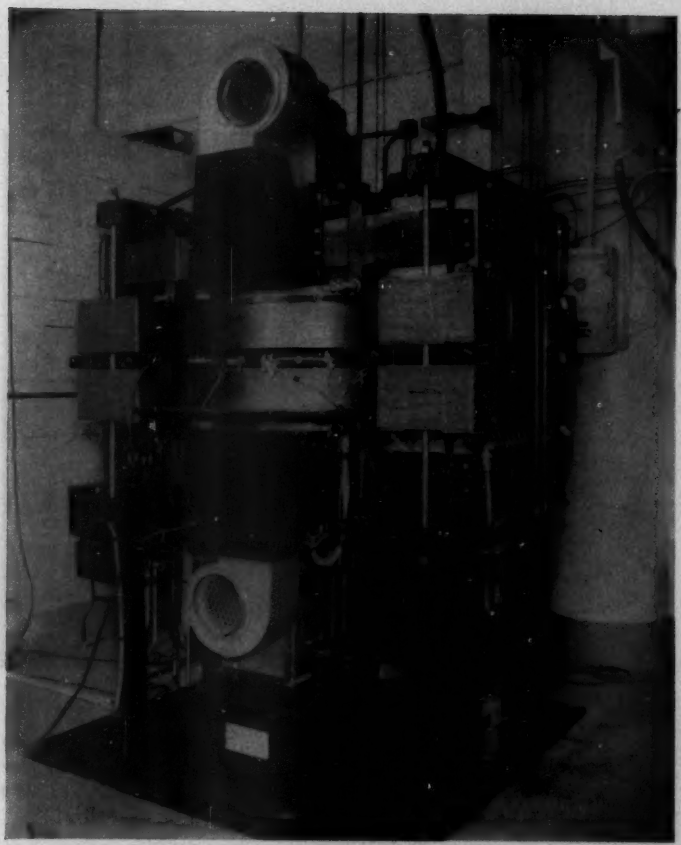


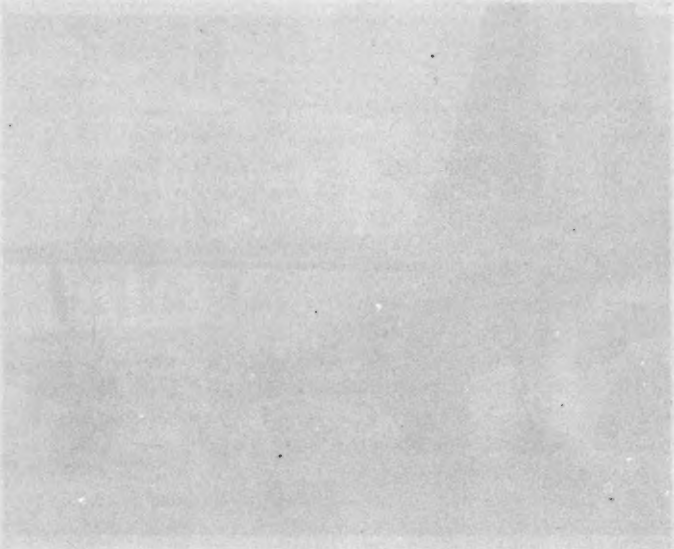
FIGURE 3.—Pictured here is the 70 million electron volt synchrotron built by the General Electric Laboratories for the Office of Naval Research. Put into action in March 1947, it is the first instrument of this type to go into operation in this country.

FIGURE 4.—A Collins Helium Cryostat, installed at the Naval Research Laboratory, shows the helium liquefier and experimental apparatus which is used in the investigation of the rectifying properties of superconductors. This is part of the equipment used in low-temperature research which has been given a new impetus in this country through efforts of the Office of Naval Research.

FIGURE 5.—Photograph taken by General Electric scientists in connection with Project Cirrus, research in cloud nucleation sponsored jointly by the Office of Naval Research and the Army Signal Corps. This photograph shows a seeded area 44 minutes after seeding, 7 April 1947, cloud altitude 6700 Ft., temperature —7°. Twenty-five lbs. of dry ice were used.

FIGURE 6.—Scientists at California Institute of Technology test the magnetic cloud chamber which they will take high above the clouds aboard a B-29 to study atom-smashing through cosmic ray research sponsored by the Office of Naval Research. Dr. Carl D. Anderson (central figure in picture), Nobel prize winner and professor of physics at Cal Tech, is heading the project.

FIGURE 7.—A technician loads equipment into a B-29 which the Office of Naval Research has fitted out as a flying laboratory to study atom-smashing through cosmic rays. The planes are based at the Naval Ordnance Test Station, Inyokern, Cal.



step is exploratory applied research, the tentative application of the new knowledge provided by basic research to the several uses for which it appears to be suited. This, the first step in development, marks off the lines of attack that hold promise from those that would be blind alleys. It is followed at once by objective applied research, which is a continuation of the work along the lines that have been identified as promising by exploratory applied research. To illustrate this phase of the work, we refer to the applied research, carried out between 1932 and 1941, by which the active agent of penicillin was isolated in stable form and efficient processes for its extraction were developed.

By the time objective applied research is well under way, the complexity that is inherent in programs that draw heavily on many different fields of scientific knowledge and technical skill asserts itself. As a result, if no special step is taken to offset the difficulties thus created, the development may slow down and require many years before success is attained. Here, however, the peculiar organizing capacity of the Western mind comes into play and a new phase of the process—programmed applied research—is initiated. This consists of organizing all available scientific and technical resources that can effectively be used to bring them to bear on the problems encountered. This function is performed for the Navy today by its research and development coordinating agencies, which are closely in touch with the scientific and technical resources of the nation as a whole, and, when funds are available, can quickly mobilize the assistance that is needed. During the war, much of the programmed applied research required by the Armed Services was provided by the National Defense Research Committee and the Committee on Medical Research of the Office of Scientific Research and Development. Probably the classical example of this phase of the research and development process is the marshaling of the country's scientific talent by the Manhattan Engineer District and OSRD for the work leading up to the production of the atomic bomb.

By the time programmed applied research is fully under way, most of the fundamental research and development problems will have been solved. The next phase of the process, accordingly, is pilot plant production, during which the development is reduced to the engineering problem of producing the device

developed in manufacturable and usable form and in such quantities as may be required. Here such problems as those of critical materials, retooling for quantity production, manufacturer's lead time, and unit cost must be satisfactorily met. For an example, turn to the problem of putting into production any new model of plane, radar, or gun. This sort of problem, fortunately, is one which America is well equipped to solve expeditiously.

Meanwhile, the seventh phase of the process—test and evaluation—has been initiated to determine the extent to which the item development meets the special naval requirements that had been established at the beginning of its development. The operational testing given any new item developed for the Navy represents this phase of the work. Test and evaluation is an integral part of the research and development process, for its findings may, and frequently do, indicate the need for introducing changes in the development itself.

Once the item has been developed and has met satisfactorily all test requirements, it is put into use by the operating units of the Navy. In wartime, for one reason or another, the demands of the Armed Services may preclude civilian use for the time being, but in war or in peace any development carried out by the Navy, the Army, or the Air Force is potentially an addition to civilian resources. Radar, for example, was used widely by the Armed Forces during the war and is now slowly coming into use for civilian purposes by the air lines and on merchant vessels.

With the introduction of an item into military use, the research and development process that produced it comes technically to an end, even though both basic and applied research may be continued in a constant effort to improve the item's quality and performance capacity.

Even in this schematic analysis of the research and development process, it is evident that the process is quite complex and that there are many points at which lack of knowledge or some other deficiency may slow down progress almost to a halt. Thus, even with the most careful organization and direction obtainable, it is impossible to know just how long it will take after a project has been initiated before a product that meets the established requirements can be turned out. Even the project officers cannot forsee accurately where major difficulties will arise; this is especially true of the basic research phases of the work. Research

and development involves the use of both science and technology, frequently calling for excursions to penetrate deeply into uncharted fields of knowledge, and often make exceptional demands on engineering skill. For these reasons, rate of progress cannot be scheduled in advance.

This brings us to our second point, namely, that we must be sufficiently acquainted with the character, complexities, and difficulties of the research and development process so that we will not expect the impossible. The development of a new item to the point where it can be used in quantity by the Fleet is a long process. Take for example the proximity fuze which played such a major role in the successful prosecution of the recent war. Its development took four years and involved what amounts essentially to development of a complete radio set, equipped with a power plant, transmitter, and receiver, all fitted into the nose of a shell. To meet Navy requirements, the fuze had to perform the following contradictions: It had to be sensitive and rapid in operation, but not subject to being triggered by the passage of other shells or by the reflection of radio waves from ground, water, and clouds. It must also be safe to handle and not subject to serious deterioration in storage. And what is more, it had to be manufactured in large quantities. That it was developed at all in four years is one of the engineering miracles of the last war. But its development involved every step from observation and basic research to test and evaluation and final production outlined above.

Because the proximity fuze was, by this near-miracle actually produced, and because other types of equipment like loran and jet propulsion did come into being, it is only too easy to conclude that research and development will solve any problem quickly and with relative ease. Given time and funds, research and development *will* solve most problems of material and technique, but they can seldom be solved on order. Long-term, carefully planned programs are the only real answer.

To put these programs into operation, ONR has been made responsible for the co-ordination of basic research, the Chief of Naval Operations for the co-ordination of development. The actual research is the responsibility of ONR and under its direction; the development of equipment is conducted by the materiel Bureaus. This organization reflects a definite trend to keep research and development activities separate. The Navy has

been early to recognize and appreciate the necessity for keeping these activities separate and it is interesting to note that industry is now organizing itself in the same way. Large industrial establishments are charging their central research laboratory with the conduct of a broad program of research, partly in support of the development progress of the manufacturing divisions and partly aimed at the acquisition of new knowledge which may lead to entirely new developments. General Electric, Westinghouse, and RCA have already done this. Others, such as Bendix Corp., have similar plans well under way.

To return to our Navy organization—the next problem is one of close coordination so that the results of research may be applied swiftly to Naval problems. To this end ONR maintains close liaison with CNO, the Bureaus, the Operation Development Force, and the Operating Fleet Units, largely by the relations of its Naval Sciences Division with the Fleet Operations Readiness Section of CNO and with the Bureaus. In addition, excellent use is made of a number of committees and boards that deal with the special research and development problems of specific fields, such as those of anti-submarine warfare and of rockets.

This over-all organization, as has just been noted, has been created to meet effectively the requirements that are inherent in research and development; that is to say, the form of the organization has been determined by the character of the job to be done. This is of special interest, for it underscores the Navy's leadership in recognizing the indispensability of basic research.

Having traced out the reasoning that lies back of the Navy's research and development activities and sketched the philosophy upon which the orgainzation has been built, we turn now to a brief review of some of the typical accomplishments of ONR in basic research during the past year. Space does not permit covering all of the fields in which progress has been made, and therefore we must be satisfied with consideration of a few in which there is considerable activity at the present time. The ones selected are weather control, cryogenics, nuclear physics, and medical sciences.

Turning to the first of these, we find that in meteorology ONR not only supports a number of basic research projects to obtain more complete data about atmospheric conditions in general, but also, in collaboration with the Signal Corps, has initiated a project

to study cloud microstructure and techniques for controlling and producing rain—and snowfall. This important project—it is called "Project Cirrus"—was assigned in February 1947 to the General Electric Research Laboratories at Schenectady by the joint action of the Signal Corps and ONR. The successes already attained have not only aroused considerable public interest in the possibility of establishing effective weather control but also have raised some interesting legal questions, ranging from the possible need for a Federal Weather Control Commission to the problem of the property rights of states and commonwealths in the atmospheric moisture above their respective domains.

Under the direction of Dr. Irving Langmuir, the General Electric scientists have sought and obtained basic information about the physics of clouds—their formation, rates of growth, and dissipation—and have given special attention to the effects of artificial nucleation of the supercooled water-droplet cloud. The program has been advanced by laboratory studies of supercooled clouds under various conditions of temperature and water content and laboratory experiments with different seeding techniques; by photographic and stereoscopic studies of clouds to determine the several factors involved in the development of their several types and also of their different structures; by a daily observation program set up to explore the possibility of inducing the development of cirrus-type clouds under clear sky conditions; and by actual seeding experiments carried out by flights of the AAF Weather Squadron assigned to the Signal Corps.

In addition to the very considerable amount of new data obtained by laboratory analysis—data which have thrown strong light on the problems of cloud physics—the seeding flights have been successful in developing effective techniques for discharging the moisture content of cirrus clouds. On 7 April 1947, for example, a seeding run was made at about 7000 feet over eastern New York and Western Connecticut; the run was about 11 miles in length, and a total of 25 pounds of dry ice, in pellets ranging from ¼ to ¾ of an inch in diameter, was used. Observation showed that the seeded area grew to a length of about 15 miles and increased in width to an average of about three miles in one-half an hour. Within the first ten minutes after seeding, some evidence of heat evolution appeared, showing that some of

the snow formed was carried above the level of the undisturbed stratus cloud.

The results of these tests indicated that, in general, a single layer of clouds in a seeded area becomes clear blue sky, as seen from the ground, within less than thirty minutes after seeding; thus within fifteen minutes after seeding it should be possible for a plane to clear a hole for itself through which it could descend without icing. Further, in many instances a plane leaving the ground should be able to fly up to the base of a stratus layer in which there is dangerous icing and, by seeding, clear a path up through which it could fly without icing. Also, within limits, the seeding technique appears to be usable for producing rain when needed on the ground, as well as to break up dangerous storms before they hit critical areas.

In passing, it should be pointed out that fine particles of silver iodide as well as of dry ice have been used successfully in cloud seeding. Effective dispensers have been developed for use with both types of material.

This work in meteorology is continuing but, for reasons that are readily apparent, in the future actual seeding experiments will be carried out over areas outside thickly-populated regions of the United States. A good beginning has been made, and every effort will be exerted to obtain the full amount of basic information needed to complete this first step toward the development of effective weather control. It is significant that throughout this phase of the work the major emphasis has been placed on basic research. As a result, when and if the time comes for inauguration of actual weather-control measures, a firm foundation of proved theory will be available to guide the practical application of the principles evolved.

As regards the work in cryogenics, which is low-temperature physics, ONR is supporting in this country a field of research conducted almost exclusively before the war in England and the Netherlands. Now why is the Navy interested in knowing more about low temperatures? Let us allow our imaginations to run over a few possible operational needs and the answer will be obvious. A situation might arise when we would want to send a submarine into an area for several weeks for the purpose of intelligence. We might require that the craft stay at periscope or "Schnorkel" depth during observations and at bottom during

the remainder of the time. Suppose now for the purposes of this discussion, that the limitation is that of sufficient oxygen and that the proposal is to equip the submarine with a plant for liquefying air through the "Schnorkel." We could store the liquid oxygen or store the oxygen gas under compression. Another situation might call for such oxygen-accumulating-gear aboard a submarine if torpedoes were to be driven by a jet fuel requiring liquid oxygen—or even the submarine itself might need bursts of speed from such a fuel. The point is that these operational needs can be met with the latest engineering designs of compact and reliable air liquefiers, and further that the production of these designs depends upon our knowledge of low temperature physics.

Liquid air, however, involves temperatures only at the beginning of the downward range of temperatures which is considered the area of study for cryogenics research. We now have some 16 laboratories in this country equipped to work with liquid hydrogen (at temperatures down to about −436°F.); seven of these have facilities to produce liquid helium (at temperatures to about −452°F.). And most of this work is being sponsored by the Office of Naval Research. Already outstanding results are coming from the investigations and physicists are getting pretty excited about the properties which some elements are exhibiting at these extremely low temperatures.

For instance it has been demonstrated that at temperatures approaching absolute zero (−494.72°F.) certain materials exhibit an intensified capacity for conducting radio waves, and also an increased diamagnetic susceptibility. Columbium nitride, for example, is normally a semi-conductor, but at about 15° K a tiny strip of it serves very well as the superconducting element of a standard Johns Hopkins superconducting infrared detector. This phenomenon of increased sensitivity at low temperatures, more and more information about which is being gained by ONR contractors, has considerable promise for use in the development of submarines and other types of naval detecting devices. If the promise is realized, the Navy soon should have available detection equipment that, in comparison with that in use today, will be so much more effective as to constitute a revolution in this phase of naval warfare.

Two ONR contractors in cryogenics work, Professor Lane of

Yale and Professor Daunt of Ohio State, have very recently discovered outstanding properties of helium 2, an element that has been known only about a year. They have found that at extremely low temperatures helium 2 behaves contrary to all existing laws of the behavior of liquids. For instance, it flows uphill. It will flow into a capillary finer than a human hair. It will flow from a cold place to a hot place. One conjecture as to the possible use of this knowledge may be that it will lead to improved methods of obtaining bearings with lower friction. Whether this will be so or not, we have at any rate added to fundamental knowledge and only time will tell what fruits this knowledge will bring.

In nuclear physics, the third field on which we are commenting here, even greater progress has been made. It is well known that the extent to which the development of the atomic bomb drew upon the accumulated basic data of nuclear physics was so great that very little untouched knowledge remained in this field at the end of the war. Also, when Japan surrendered, the great organization of nuclear physicists that had been built up by the Manhattan Engineer District with the assistance of OSRD was torn down, and many of the leading scientists returned to their own laboratories. The Manhattan District and, later, its successor, the Atomic Energy Commission, were not prepared to support basic unclassified research. The Office of Naval Research stepped into this picture and provided funds not only for the modification and reconversion of old nuclear research equipment, but also for new construction required for active prosecution of basic research. Thus, the continuation of this country's leading nuclear research development and engineering was assured. Today the Office of Naval Research spends about six million dollars a year on nuclear physics and is supporting 90 per cent of the current fundamental research in this very important field.

Possibly the major concentration of attention has been on the study of the composition of the nucleus and of the nature of fundamental particles. At Washington University in St. Louis, for example, research has been conducted on the photo-disintegration of the deuteron and the beryllium nucleus by gamma rays. The Massachusetts Institute of Technology has studied the reaction taking place when two observable particles are ejected from a nucleus as the result of bombardment.

One of the major forms of support ONR has given this phase of research in nuclear physics has been in the design and construction of improved cyclotrons, synchrotrons, and other particle accelerators by various laboratories throughout the country. Under contracts with ONR, the cyclotrons of Washington University at St. Louis and Princeton have been greatly improved in design and performance; new cyclotrons, several of them of novel design and greatly increased size, have been carried far toward completion at Columbia, Harvard, Rochester, and Carnegie Institute of Technology. In addition, the 300 million electron volt synchrotron at Massachusetts Institute of Technology and the 70 million electron volt synchrotron at General Electric Laboratories have been completed; the latter, which was put into action in March 1947, was the first instrument of this type to go into operation in this country. Moreover, the design of a 300 million electron volt non-ferrous synchrotron was completed by the General Electric Company. In like manner, a number of new-type accelerators, ranging from a small (4 million electron volt) microwave linear device at Purdue to the proposed billion electron linear accelerator at Stanford have been designed and construction initiated.

The relatively great operational capacity of these new instruments is opening new avenues of research in nuclear physics, and promises to produce in the near future a wealth of new data that will be of incalculable value in the development of atomic energy devices.

A second major field of research in nuclear physics during the past year has been the study of cosmic rays, in particular to determine the character of mesotrons. The creation and decay of mesotrons, which appear to take place only at relatively high altitudes (that is, above 20,000 feet above sea level) are as yet little understood, but they seem to represent highly important conversions that take place between matter and energy. Better understanding of this process may provide clues to future sources of nuclear power.

In conjunction with this research, high altitude research with B-29's and balloons has provided a considerable amount of basic information, which since has been added to by the work of several contractors, among which the California Institute of Technology has been outstanding. New counters have been developed, and

valuable progress has been made in determining the character and behavior of mesotrons.

In still another field of nuclear physics, work on radio isotopes has produced very encouraging results.

In all, much progress was made under ONR sponsorship during 1946-47 in the field of nuclear physics. In particular, not only have great advances been attained in such basic researches as those on the structure of the nucleus and the character of mesotrons, but very valuable and highly productive research equipment has been designed and put into operation. It is reasonable to anticipate that, as a result of this progress, nuclear physics will make great strides in the very near future.

Here again, as we have just seen, the emphasis of the work sponsored by ONR has been almost exclusively on basic research.

In medical sciences research, the fruits have been particularly rich. A new method of arterial grafting has been developed which makes possible for the first time the permanent repair of injured sections of arteries without restriction of blood flow. Also under ONR contract a neurosurgeon has developed a successful technique for elongating nerves and thus allowing reinervation of damaged limbs. A successful method has also been devised of replacement of diseased bones with inactive plastic substances, thus enabling the early rehabilitation of the patients, while the material is being replaced by normal tissue. The use of this material for the devising of an artificial joint seems to be a definite possibility. Progress has been made in the field of microbiology and greater knowledge has been made available concerning new antibiotics such as streptomycin. It is hoped that increased medical knowledge will help maintain the Navy complement at maximum effectiveness, and in fact bring greater benefits in health to the nation as a whole.

In addition to its primary responsibility—the co-ordination and direction of Naval research—ONR is also required to advise the Chief of Naval Operations of the developments that may be expected to evolve from current research, to inform him of the trends and potentialities of development that research is making possible, and to collaborate with him and the Chiefs of the Bureaus in the formulation and modification of the Navy's principal development programs. As has already been suggested, most of this work is carried out for ONR by its Naval Sciences

Division. This Division has adopted a system of program analysis as a means of performing these duties most effectively.

Program analysis focuses attention at once on the critical technical difficulties that must be overcome before a problem can be completely solved, and thereby makes possible full co-ordination of research and development activities with the various military requirements. Furthermore, it indicates the extent to which partial solutions of different phases of the problem will give to the Fleet equipment and techniques that it can use in improving its operational performance in the field in which the work is being done.

In this way ONR advises the Chief of Naval Operations on the trends and potentialities of research and development as they affect the special problems of naval operations with which he is concerned. This makes possible increased effectiveness in co-ordinating the resources available in solving these problems. As such programming is applied to more and more fields of naval warfare, the resulting improvement in the co-ordination of research and development activities should produce considerable savings in expenditures, and bring about greater progress on the budgets provided.

However, these are just goals. We do not have the problem fully under control. We are continually studying means for greater improvement in the coordination of research and development activities, and naturally in the Navy which is the largest engineering and technical organization in the world, the size of the problem is great.

In summarizing these comments on the actual work supported, co-ordinated, and carried out by ONR, we come to our *third point, namely, that the Navy, through ONR, not only is in the lead in recognizing the indispensability of basic research to continued development but also is taking the steps necessary to the organization and implementation of such a program.*

Throughout this discussion, the principle that science is an essential element of naval research and development has been emphasized. In conclusion, the following points regarding its role in the research and development process are submitted to summarize these remarks.

First of all is the obvious point that, *unless* basic research is strongly supported at a high level of quality and productivity,

development in the long run will suffer and in some fields may even come to a standstill. Without a continuing expansion of scientific knowledge, development is bound to slow down.

In keeping with this point, it should be added that the experience of World War II demonstrated that, when relatively complete scientific knowledge is available in a given field, development is reasonably easy, rapid, and certain. Also, that the principle already referred to, namely that research is revolutionary,—requires naval authorities to support scientific research in order that their special sources of scientific knowledge will not fall behind those of other powers in the competition for superiority of materiel. It is in these respects that programmed applied research and co-ordinated basic research play their parts most effectively.

Secondly, it is indicated that basic research must be as free as possible from the limitations imposed by security classification. A free interchange of fundamental information is necessary to genuine scientific inquiry. As soon as basic research and exploratory applied research lead to applications that should be classified for security reasons, those lines of attack can be closed off from public knowledge by appropriate security action, but until then the work should be free. Basic research is an inquiry into the unknown and, to be successful, must be freely prosecuted by the best trained minds that become interested in the problem. It cannot profitably be reduced to administrative control. For its part, ONR prefers to bet on the man. Having sought and found the individual best qualified to carry on certain research, it gives him the support he requires and assists him when and where necessary, relying upon his knowledge, judgment, and skill to carry through to a successful completion of the job.

Thirdly, as soon as basic and exploratory applied research have pointed the way and the scientific and technical intricacies of the problem begin to pile up, programmed applied research on the scale demanded by the situation becomes imperative in order to speed up the development.

In the fourth place—and this is closely related to the need for programming—sufficient funds must be provided to make possible the effective prosecution of the work. Compare the slow progress in penicillin research made between 1928 and 1933, when practically no outside funds were available, with the

relatively rapid advancement of 1938-1942, when the Governments of the United States and the United Kingdom began to give a helping hand. Frequently the funds needed will be very large but, if the research and development programs are to be carried to successful conclusions, they must be provided.

Finally, many of the products of naval and military research and development become in relatively short order contributions to civilian life. In this respect, the funds invested in these activities serve a double purpose; they contribute to national security and also add to the domestic welfare and prosperity of the nation itself.

Fleet Admiral Nimitz, upon his departure from the Navy, recently pointed out that the chief objective in an era of atomic warfare will be to keep the threat of atomic attack—by air or ship—as far from this hemisphere as possible. The Navy will provide this country's first line of defense on, under, and above the sea in the grim nuclear tomorrow. "The Navy of the future," he indicated, "will be capable of launching missiles from surface vessels and submarines, and of delivering atomic bombs from carrier-based planes. Vigilant research is constantly developing and adding to these means." "Vigilant research," our leader called for, and in keeping with the traditions of the Navy, we are complying.

NOTES.

INDEX TO NOTES.

CANONS OF ETHICS FOR ENGINEERS.

WELDING TECHNIQUES FOR CAST IRON.
—The Welding Journal, January, 1948

BURNING BOILER FUELS IN MARINE DIESEL ENGINES.
—The British Motor Ship, January, 1948.

AMERICA'S HIGHEST-POWERED MOTORSHIP.
—Motorship, September, 1947.

SELF-ALIGNING BEARINGS FOR PROPELLER SHAFTING.
—Shipbuilder and Marine Engine Builder, August, 1947.

STRUCTURAL FAILURE OF A RIVETED SHIP.
—The Welding Journal, January, 1948.

ALUMINUM ALLOY BEARINGS IN DIESEL ENGINES.
—The British Motor Ship, September, 1948.

THE "WALKING BARGE."
—The Military Engineer, January, 1948.

SHEARING STRESS DISTRIBUTION IN BOX GIRDERS WITH MULTIPLE WEBS.
—Proceedings American Society of Civil Engineers, February, 1948.

On October 27, 1947, The Engineers' Council for Professional Development adopted The Canons of Ethics for Engineers printed below. This code of ethics was developed jointly over a period of years by official representatives of the ASME, ASCE, AIEE, AIMME, AICE, SPEE, Engineering Institute of Canada, and National Council of State Boards of Engineering Examiners.

CANONS OF ETHICS FOR ENGINEERS.

FOREWORD.

Honesty, justice and courtesy form a moral philosophy which ,associated with mutual interest between men, constitutes the foundation of ethics. The engineer should recognize such a standard, not in passive observance, but as a set of dynamic principles guiding his conduct and way of life. It is his duty to practice his profession according to those Canons of Ethics.

As the keystone of professional conduct is integrity, the engineer will discharge his duties with fidelity to the public, his employers and clients, and with fairness and impartiality to all. It is his duty to interest himself in public welfare, and to be ready to apply his special knowledge for the benefit of mankind. He should uphold the honor and dignity of his profession and also avoid association with any enterprise of questionable character. In his dealings with fellow engineers he should be fair and tolerant.

PROFESSIONAL LIFE.

Sec. 1. The engineer will co-operate in extending the effectiveness of the engineering profession by interchanging information and experience with other engineers and students and by contributing to the work of engineering societies, schools and the scientific and engineering press.

Sec. 2. He will not advertise his work or merit in a self-laudatory manner, and he will avoid all conduct or practice likely to discredit or do injury to the dignity and honor of his profession.

RELATIONS WITH THE PUBLIC.

Sec. 3. The engineer will endeavor to extend public knowledge of engineering, and will discourage the spreading of untrue, unfair, and exaggerated statements regarding engineering.

Sec. 4. He will have due regard for the safety of life and health of the public and employees who may be affected by the work for which he is responsible.

Sec. 5. He will express an opinion only when it is founded on adequate knowledge and honest conviction while he is serving as a witness before a court, commission or other tribunal.

Sec. 6. He will not issue ex parte statements, criticisms or arguments on matters connected with public policy which are inspired or paid for by private interests, unless he indicates on whose behalf he is making the statement.

Sec. 7. He will refrain from expressing publicly an opinion on an engineering subject unless he is informed as to the facts relating thereto.

RELATIONS WITH CLIENTS AND EMPLOYERS.

Sec. 8. The engineer will act in professional matters for each client or employer as a faithful agent or trustee.

Sec. 9. He will act with fairness and justice between his client or employer and the contractor when dealing with contracts.

Sec. 10. He will make his status clear to his client or employer before undertaking an engagement if he may be called upon to decide on the use of inventions, apparatus, or any other thing in which he may have a financial interest.

Sec. 11. He will guard against conditions that are dangerous or threatening to life, limb or property on work for which he is responsible, or, if he is not responsible, will promptly call such conditions to the attention of those who are responsible.

Sec. 12. He will present clearly the consequences to be expected from deviations proposed if his engineering judgment is overruled by non-technical authority in cases where he is responsible for the technical adequacy of engineering work.

Sec. 13. He will engage, or advise his client or employer to engage, and he will cooperate with, other experts and specialists whenever the client's or employer's interests are best served by such service.

Sec. 14. He will disclose no information concerning the business affairs or technical processes of clients or employers without their consent.

Sec. 15. He will not accept compensation, financial or otherwise, from more than one interested party for the same service, or for services pertaining to the same work, without the consent of all interested parties.

Sec. 16. He will not accept commissions or allowances, directly or indirectly, from contractors or other parties dealing with his client or employer in connection with work for which he is responsible.

Sec. 17. He will not be financially interested in the bids as or of a contractor on competitive work for which he is employed as an engineer unless he has the consent of his client or employer.

Sec. 18. He will promptly disclose to his client or employer any interest in a business which may compete with or affect the business of his client or employer. He will not allow an interest in any business to affect his decision regarding engineering work for which he is employed, or which he may be called upon to perform.

Relations with Engineers.

Sec. 19. The engineer will endeavor to protect the engineering profession collectively and individually from misrepresentation and misunderstanding.

Sec. 20. He will take care that credit for engineering work is given to those to whom credit is properly due.

Sec. 21. He will uphold the principle of appropriate and adequate compensation for those engaged in engineering work, including those in subordinate capacities, as being in the public interest and maintaining the standards of the profession.

Sec. 22. He will endeavor to provide opportunity for the professional development and advancement of engineers in his employ.

Sec. 23. He will not directly or indirectly injure the professional reputation, prospects or practice of another engineer. However, if he considers that an engineer is guilty of unethical, illegal or unfair practice, he will present the information to the proper authority for action.

Sec. 24. He will exercise due restraint in criticizing another engineer's work in public, recognizing the fact that the engineering societies and the engineering press provide the proper forum for technical discussions and criticism.

Sec. 25. He will not try to supplant another engineer in a particular employment after becoming aware that definite steps have been taken toward the other's employment.

Sec. 26. He will not compete with another engineer on the basis of charges for work by underbidding, through reducing his normal fees after having been informed of the charges named by the other.

Sec. 27. He will not use the advantages of a salaried position to compete unfairly with another engineer.

Sec. 28. He will not become associated in responsibility for work with engineers who do not conform to ethical practices.

WELDING TECHNIQUES FOR CAST IRON.

The following article by Mr. T. E. Kihlgren of the International Nickel Co. is reprinted from "The Welding Journal" for January, 1948.

Introduction.

There are various mechanical methods and fusion welding and brazing processes which may be used to reclaim defective castings or to repair cast-iron parts which have failed in service. No doubt there are instances where one specific method or process is peculiarly fitted to meet the requirements, and there are unquestionably many occasions when the fields of applicability of the various methods overlap or even coincide.

No attempt will be made to evaluate these various methods of salvage or repair. It is proposed rather to confine our discussion of techniques for welding cast iron to those techniques developed and found useful in conjunction with arc welding, using nonferrous electrodes of the nickel or nickel-copper type. Field and laboratory experiences discussed herein are based primarily on arc welding using nickel electrodes.

Purpose of Arc Welding.

There are broadly three main categories into which arc welding of cast iron may fall: (1) salvaging of defective castings in the foundry, (2) maintenance and repair welding and (3) fabrication welding.

The contribution made by the welding industry to the gray iron foundry has been one of constantly growing importance, and, today, salvage of defective castings by welding is widely and successfully practiced. The increasing publicity being given by the foundry industry to welding as a production aid will not only result in more widespread application of cast-iron welding, but should also aid in preventing its misapplication.

Maintenance and repair welding of cast-iron parts which have failed in service has been practiced for many years using brazing, oxyacetylene or metallic arc processes. The trend in the last several years has been toward increasing application of arc welding, a trend accelerated by the war and by improvements in available electrodes.

Fabrication welding of cast iron is at a somewhat embryonic stage and its potentialities have yet to be realized. The component parts of the weldment may consist of cast iron only, or of castings and forgings or wrought parts of other compatible metals. In some instances, casting difficulties inherent in a casting design may be overcome by using a weldment of two or more castings. Welding may also be used to join cast iron to other metals, involving such operations as the attaching of steel brackets or fittings to gray iron castings. It is obvious that some caution should be exercised in this third phase of welding, since there is a relatively small background of experience upon which the welding engineer can draw.

It is proposed in the following discussion to present a few techniques which have been found useful at the laboratory and in the field. A few examples will also be given of each of the three categories into which we have arbitrarily classified welding of cast iron.

Welding with Nonferrous Electrodes.

Although arc welding of cast iron is done with both ferrous and nonferrous electrodes, the latter possess the advantage of producing machinable welds even when no preheat is used. While by far the largest portion of arc welding with nonferrous electrodes is done without preheat, there are circumstances under which a moderate amount of preheat is either helpful or advisable.

The use of nickel-copper electrodes for producing machinable welds on cast iron has been practiced for at least about 30 years. During this period, improvements were made in the flux coating to yield electrodes which possessed good arc and slag characteristics and deposited welds quite satisfactory with respect to weld porosity considerations. The nickel-copper type of electrode is used in considerable quantities both here and abroad.

Observations at the laboratory, made over a number of years, had indicated commercially available nickel electrodes to have technical advantages over nickel-copper electrodes, particularly with respect to weld hot cracking tendencies. However, these nickel electrodes had not been designed expressly

for cast-iron welding, and attention was directed toward the development of a nickel electrode having suitable arc and slag characteristics when applied to cast iron, and operable on either a.-c. or d.-c. current.

Such an electrode was developed as a result of the laboratory studies and ultimately put into production. In connection with this work, a small laboratory extrusion press was used for preparation of moderate quantities of experimental flux-coated electrodes. It permits preparation of modest quantities of electrode ranging from 0.075 to 5/16 in. in diameter. The flux-coating thickness for a given diameter can also be varied over a fairly wide range.

The nickel electrode which was thus developed and upon which most of the welding techniques to be discussed were based can be briefly described. It consists simply of a nickel core wire coated with a carboniferous lime-base flux containing a moderate amount of metallic reducing agents. Sufficient carbon is introduced through the flux coating to obtain graphitic carbon in considerable amounts in the fusion, resulting in improved solidification characteristics as well as a structure imparting a degree of machinability similar to that of the cast iron.

As is well known, there are a number of nickel electrodes now commercially available. Many of the observations made and techniques discussed in the balance of this discussion will be applicable to them as well as to the various nickel-copper electrodes. However, whatever nonferrous electrode is used should be employed according to the manufacturer's instructions, which should indicate the optimum condition of operation for that particular electrode.

Influence of Welding Technique on Machinability.

While the degree of machinability will vary somewhat with the hardness and structure of the deposit, any weld made on cast iron using nonferrous electrodes is machinable as far as the weld deposit itself is concerned. The condition of the cast iron in the heat-affected zone and at the fusion line, therefore, determines the machinability of the welded joint. The structure of the cast iron in these areas is primarily determined by the composition of the cast iron and by the cooling rate following welding.

Figure 1 shows the machined cross sections of welds made on 1-in. thick cast-iron plate without preheat using 3/16-, 5/32-, and 1/8-in. electrodes. The cuts were made in a shaper at 33 strokes per minute, 0.020 in. feed and 0.015 in. depth of cut. The effect of the superimposed passes in tempering the heat-affected zone adjacent to previous beads may be observed from the figure, the center of the joint showing practical absence of a hard zone. The annealing effect of superimposed passes can be employed to reduce or eliminate hard zones in the cast iron adjacent to the weld, at the surface of the casting. This can be done simply by completing the joint flush to the surface, cooling to about 70-200° F. and then using cover passes, confined entirely to the weld, to reheat and temper the heat-affected zone. Figure 2 shows the effect of such a procedure. The top plate has not been subject to the "cover pass" tempering bead technique while the two below were so treated, the second with a single and the third with a double cover bead. This technique is recommended by several manufacturers of nonferrous welding electrodes.

Occasionally it is necessary to repair surface defects on raised areas such as bosses which must subsequently be machined. Sometimes the mistake is made of merely chipping the defects, welding, and then machining back to the desired level. It is preferable to prepare the area for repair by machining to a little below the desired final dimension, and then building up the surface and machining to the desired level. By such a procedure, all machining is completely confined to the weld deposit. Figure 3 schematically shows "wrong" and "right" methods of building up boss sections.

Preheating and Postheating.

In some instances, where machining requirements are especially severe, the use of a preheat may occasionally be indicated, a preheat of as much as

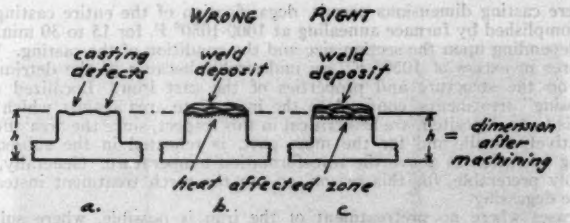


FIGURE 3.—RECOMMENDED TECHNIQUE FOR BUILDING UP BOSS TO OBTAIN MAXIMUM MACHINABILITY.

450-600° F. sometimes being employed. A preheat may be particularly useful on a light section not lending itself well to the "cover" or annealing pass technique. It should not be overlooked that post annealing may occasionally be usefully applied. A postheat sufficient to temper martensite may be helpful in reducing the hardness of the heat-affected zone and thus increasing the ability to resist service stresses. This is not common practice, but probably merits consideration where conditions warrant. Postheating may be accomplished with a torch if necessary.

METHOD OF MAKING MOLD REPAIR.

While acceptable repairs can be made on mold surfaces (glass molds, for example) by conventional methods of chipping and welding, one mold manufacturer uses a more effective method which is depicted in Figure 4. The

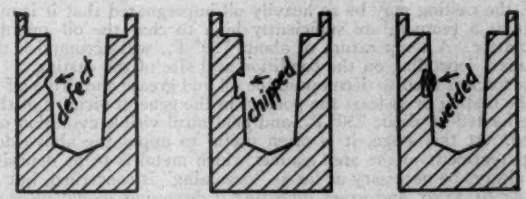


FIGURE 4.—METHOD USED BY MOLD MANUFACTURER TO "LOCK" WELD REPAIR.

center figure shows the defect undercut by chipping prior to welding, and the right-hand figure shows the completed repair. With this method, the deposit is "locked in," minimizing danger of the deposit spalling due to failure in the heat-affected zone of the cast iron.

DEGASSING OF CASTINGS.

Occasionally, castings are encountered which contain dissolved gas. While such castings may be perfectly satisfactory from a service viewpoint, attempts to repair such castings by welding lead to weld porosity, due to release of the gas. This difficulty can be overcome by prior subjection of the casting to a simple degassing treatment. The treatment consists of torch heating the *immediate area* against which metal is to be deposited to a dull red, holding briefly and allowing to cool. Figure 5 demonstrates the efficacy of the degassing treatment. Both butt joints were made on the same iron (in this instance a "bath tub" iron of the analysis: 3.40 TC, 2.40 Si, 0.65 Mn, 0.75 P, 0.12 S). The joint on the left was made without prior treatment of the iron, and that on the right after degassing.

Where casting dimensions permit, degasification of the entire casting can be accomplished by furnace annealing at 1000-1050° F. for 15 to 30 min., the time depending upon the section size and the condition of the casting. Temperatures in excess of 1050° F. are undesirable because of the detrimental effect on the structure and properties of the cast iron. Localized torch "degassing" treatments, confined to the immediate area against which weld metal is to be deposited, are less critical in this respect, since the area affected is relatively small, and for the most part, is reheated in the subsequent welding operation to above the transformation temperature. Generally, it is probably preferable, for this reason, to use the torch treatment instead of furnace degassing.

In cases where no pretreatment of the iron is possible, where suitable facilities are not available, or where the requirements are not strict, the first passes can be used to drive off the gas in the vicinity of the weld, the bead chipped and welding continued. This is a less satisfactory procedure than torch degassing, but circumstances may make it acceptable.

It should, of course, be recognized that degassing treatments are not needed in a majority of salvage and repair welding applications. However, the above procedure should be found effective in overcoming such difficulties when encountered.

Oil Impregnated Castings.

Castings which have failed in service, in some cases, will have been used under conditions causing them to be saturated with oil and grease, which must be removed before any satisfactory welding can be accomplished. The use of a suitable solvent such as carbon tetrachloride, sometimes in conjunction with cleaning by steam, may often be sufficient where the penetration is superficial.

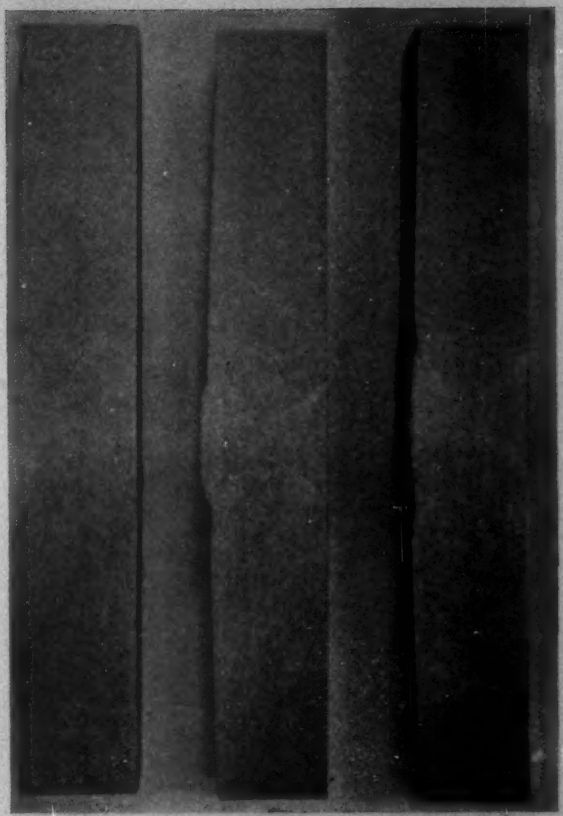
In some instances, particularly where the cast iron has a rather open structure, the casting may be so heavily oil impregnated that it is necessary to heat it at a temperature sufficiently high to char the oil and drive off volatile matter. A temperature of about 750° F., will accomplish this, the time required depending on the condition and size of the casting.

When a torch is used to decompose the oil and grease and drive off volatile matter, the casting or at least the portion in the general vicinity of the weld, should be heated at about 750° F., and held until visible evolution of fumes has ceased. At this stage, it is often useful to apply the above described degassing treatment to the area against which metal is to be deposited. It may not always be necessary as far as "degassing" is concerned, but it takes little longer to do so, and saves reheating if degassing should ultimately be found desirable.

Welding of Cast Iron to Other Metals.

There have been a number of applications in the field where the welding of steel to cast iron using nickel electrodes has been very successfully accomplished. Occasionally it may also be necessary to join cast iron to other metals. In order to get some idea of the possibilities using nickel electrodes, the following single-vee 60° included angle butt joints were made on 3/16-in. thick plates:

No.	Plate	Electrode	Plate
1	Cast iron	Nickel	Mild steel
2	Cast iron	Nickel	18–8 stainless
3	Cast iron	Nickel	Monel
4	Cast iron	Nickel	Nickel
5	Cast iron	Nickel	Inconel



$^{3}/_{16}$ IN DIAM. ELECTRODE | $^{5}/_{32}$ IN. DIAM. ELECTRODE | $^{1}/_{8}$ IN. DIAM. ELECTRODE

FIGURE 1.—MACHINED CROSS SECTIONS OF WELDS OF 1-IN. PLATE MADE WITHOUT PREHEAT.

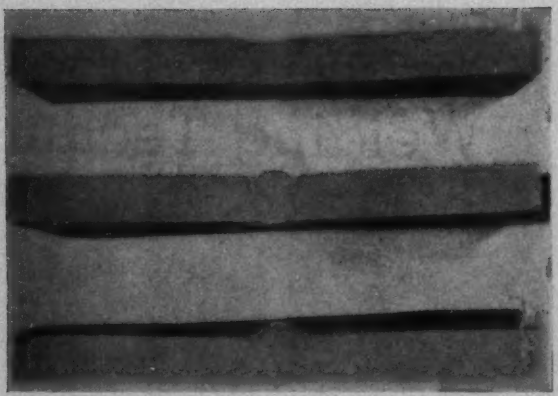


FIGURE 2.—MACHINED CROSS SECTIONS OF WELDS ON ½-IN. CAST-IRON PLATE SHOWING ANNEALING EFFECT OF COVER PASSES.

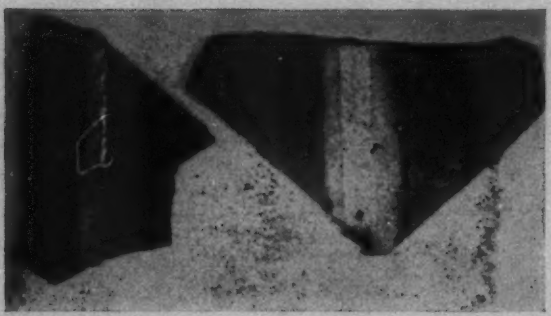


FIGURE 5.—EFFECT OF TORCH DEGASSING TREATMENT APPLIED TO HIGH GAS CONTENT CAST IRON, PRIOR TO WELDING.

FIGURE 6.—CAST IRON WELDED TO VARIOUS METALS WITH NICKEL ELECTRODE.

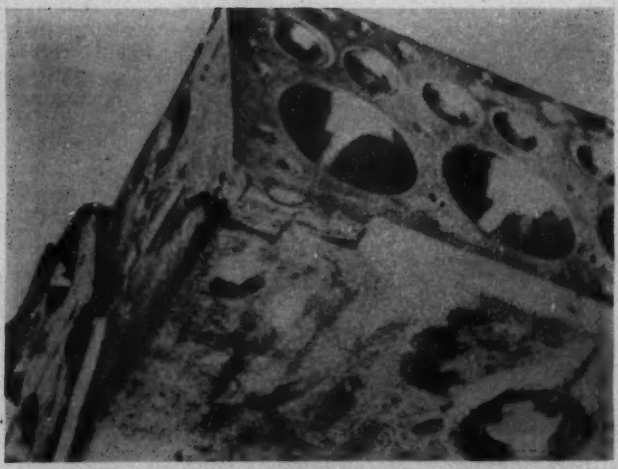


FIGURE 7.—ENGINE BLOCK WITH CASTING DEFECT AT CORNER.

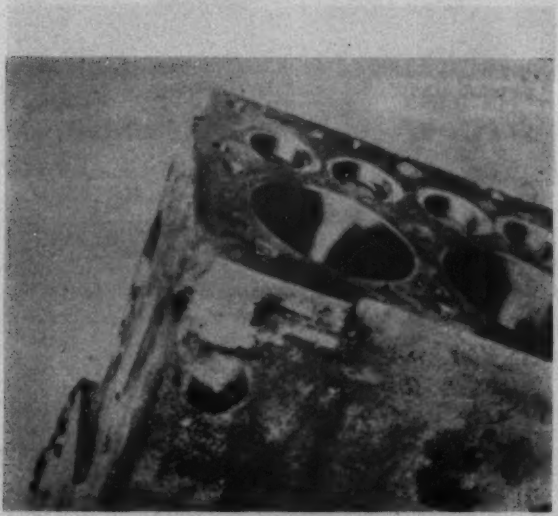


FIGURE 8.—ENGINE BLOCK AFTER REPAIR WELDING WITH METALLIC ARC WITHOUT PREHEAT.

FIGURE 9.—GEAR WHEEL ON 60-TON PRESS PREHEAT 350-400° F.

FIGURE 10.—75-TON STAMPING PRESS YOKE PREHEAT 350-400° F.

FIGURE 11.—COMPRESSION CHAMBER OF PRESSURE DIE CASTING MACHINE REPAIRED BY WELDING STEEL TOP TO CAST-IRON CYLINDER WITH NICKEL ELECTRODES.

FIGURE 12.—FABRICATED AERODYNE FAN WITH CAST-IRON VANES ARC WELDED TO INNER AND OUTER RINGS OF HEAVY GAGE STEEL SHEET.

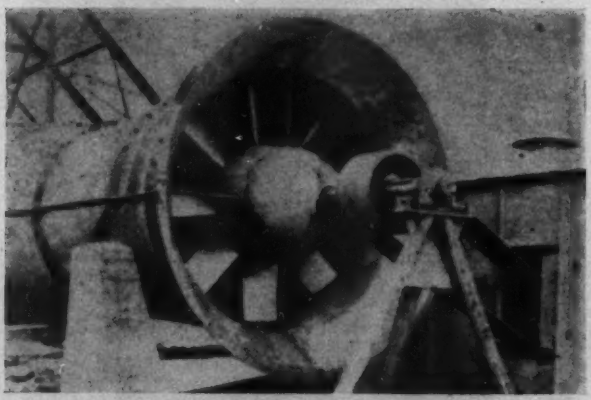


FIGURE 13.—FAN INSTALLED IN AIR SHAFT OF MINE.

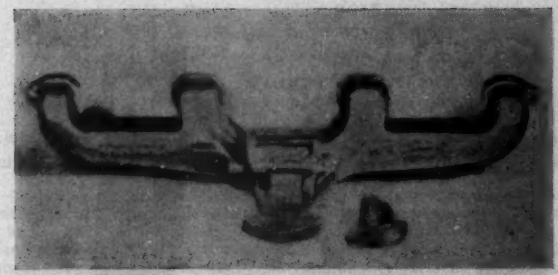


FIGURE 14.—MANIFOLD BEFORE ATTACHING SECONDARY PORT SEEN IN FOREGROUND.

FIGURE 15.—MANIFOLD SHOWING SECONDARY PORT WELDED TO CASTING.

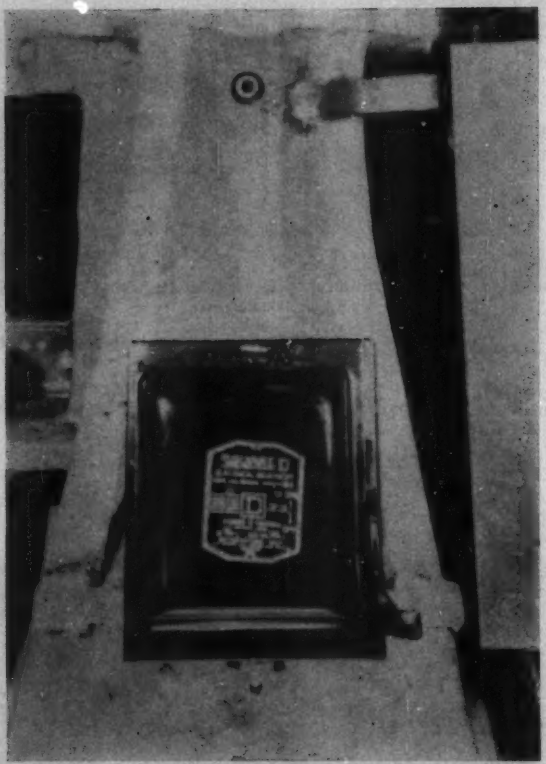


FIGURE 16.—SHOWING METHOD OF ATTACHING SWITCH BOX SUPPORTING BARS OF MILD STEEL TO CAST-IRON BASE.

No difficulty was encountered in securing a good bond in all cases. A Vickers hardness survey (50 kg. load, two-thirds objective) showed the following:

	Plate	Weld	Plate
1	175	171	138
2	181	162	168
3	179	160	146
4	176	173	133
5	177	183	166

The above joints are illustrated in the composite photograph of Figure 6.

Salvage Welding in the Iron Foundry.

Welding is an invaluable tool in the foundry. Defects due to misruns, sand wash, cold shuts, external shrinks, pattern errors, etc., often can be satisfactorily corrected. Judgment must be exercised as to the feasibility of reclamation and whether or not it is economically justified. Such factors as size, shape and initial cost of the castings, and the magnitude, nature and location of the defects enter into the decision on whether to scrap or to attempt salvage.

Figure 7 shows an engine block, partially machined which contained a defect in the corner. Figure 8 shows the same area after arc welding without preheat, welding intermittently in order to keep the block temperature as low as possible. This is important in a repair of this type, especially when made after partial machining, as excessive heat input may cause distortion and get the cylinder bore out of round.

Maintenance and Repair Welding.

This is the most common and oldest field of application of nonferrous electrodes. Speed of repair is often of paramount importance and arc welding may then be the only feasible method. Massive equipment can usually be repaired by arc welding with nonferrous electrodes without the need for dismantling, thus saving several days of labor. Figure 9 shows a fly-wheel on a 60-ton press and Figure 10 a 75-ton stamping press bracket or yoke, both welded with nickel electrodes. A 350-400° F. preheat applied to the general vicinity of the area to be welded was used in both repairs.

Fabrication Welding.

As we have mentioned before, this field of welding has not yet been fully explored. Fabrication welding should be undertaken with due recognition of the nature of the material being welded. Under conditions involving moderate sections and joints which are not subjected to excessive restraint, welding fabrication involving castings only, or cast and wrought parts, can often be satisfactorily accomplished.

Figure 11 illustrates a repair weld, but one which nevertheless is typical of problems involved in fabrication welding. In this application, the pressure chamber of the zinc base die casting machine was originally all made of cast iron. Due to misalignment, the cast-iron top fractured when struck in impact by the plunger. After a number of failures and repairs, it was decided to repair the pressure chamber by cutting off the square top and welding a mild steel plate to the cylinder. This joint, involving steel to cast iron, was made successfully with nickel electrodes.

Figure 12 shows a fabricated aerodyne fan with cast-iron vanes arc welded to inner and outer rings of heavy gage steel sheet. Figure 13 shows such a fan installed in the air shaft of a mine. The previous procedure on this job involved the drilling of 60 holes in the vanes, the cutting to size of 60 pieces of steel, the drilling of 120 holes in the steel strips and the setting of 60 rivets. One bead was deposited around each corner of each vane, without the use of preheat.

Figure 14 depicts a cast-iron automotive manifold, as purchased from a foundry. The company concerned wish to cut a hole in the manifold and attach a secondary port to it. The separate casting used for the latter port is shown in the foreground. Figure 15 is a photograph of the casting after the welding had been completed. The fabricator indicated the following procedure: inspection of parts, preheat of manifold to 450° F., port set in fixture and mounted on manifold, tack welding of port in three places, removal of fixture followed by continuous welding, and lastly, a postheat to 450° F., removal of slag and air cooling.

An a.-c. welder was used with a reported welding current of 160 amp., using $\frac{1}{8}$-in. diameter electrodes. The position of the port was maintained accurately at a 3½° angle, ±½° and all other dimensions were held within ±0.003 in. Boring of the port was easily accomplished.

Figure 16 illustrates an application of welding of steel to cast iron, the steel supporting bars on which the switch box is mounted being arc welded to the cast-iron base of the stock turret. Formerly the bars were bolted to the base, installation of the box involving chipping of the casting at four positions, and drilling and tapping of bolt holes. The present procedure consists of single pass welds, using $\frac{5}{32}$-in. electrodes, without preheat. The electrode is also used to build up the surface where the top bar is attached, in order to bring both supporting bars into the same vertical plane.

These illustrations of fabrication welding should serve to indicate possibilities that exist in this relatively unexplored field, where attractive savings in time and labor often can be made. Again, however, it is well to emphasize that such applications should be approached with due respect for the characteristics of cast iron, and the results of initial fabrication attempts should be checked with some care.

In closing, it is recognized that arc welding with nonferrous electrodes is merely one phase of fusion welding of cast iron, and a discussion of welding techniques for cast iron should properly include all the commonly employed methods. It is also apparent that there is much more yet to be said with respect to techniques of cast-iron welding, even when confined to nonferrous electrodes. However, it is hoped that this discussion will have served to stimulate further interest and bring out new ideas in this important field of application of welding.

BURNING BOILER FUELS IN MARINE DIESEL ENGINES.

The following extracts from a paper read before the Institute of Marine Engineers by Mr. John Lamb are reprinted from "The British Motorship" for January, 1948.

In composition and physical properties, the fuel required by motor ships varies only slightly, but in fuels commonly used in steamships these factors vary more widely, as will be seen from the following typical specifications:—

From the differences in costs per ton it will be seen that if a motor ship could be made to operate on any of the boiler fuels listed above, which are available on most routes followed by steamships, the saving in the fuel bill would be substantial. Actually, the annual saving in the case of a 4000 Ihp. installation having a fuel consumption of 0.3 lb. per Ihp.-hr. operating for 300 days yearly and lifting bunkers at Dutch West Indian ports would be:—

	£
No. 1 boiler fuel	3340
No. 2 boiler fuel	4240
No. 3 boiler fuel	4870
No. 4 boiler fuel	5030

Table 1.

	Diesel Fuel	Boiler Fuels			
		1	2	3	4
Specific Gravity @ 60°F.	0·896	0·94	0·960	0·980	0·99
Viscosity Red. I @ 100°F. Secs. ...	43"	430"	1480"	3400"	6400"
Flash Point P.M. Closed	188°F	200°F.	207°F.	200°F.	196°F.
Pour Point *below* ...	20°F	20°F	20°F.	20°F.	20°F.
Ash Content ...	0·02%	0·05%	0·05%	0·07%	0·06%
Sulphur Content ...	1·1%	1·9%	2·3%	2·7%	2·8%
Water Content ...	0·03%	0·05%	0·05%	0·05%	0·05%
Carbon Residue Conradson ...	1·25%	8%	10%	14%	14%
Sediment by Extraction	0·03%	0·03%	0·03%	0·04%	0·08%
Calorific Value B.T.U. s lb. ...	18,950	18,750	18,600	18,400	18,300
Price per Ton...	83/-	65/8	61/-	57/9	56/11

With higher powers the saving would be proportionately increased, and that for an 8000 Ihp. installation employing even No. 3 boiler fuel would be particularly attractive. In calculating the above savings no allowance has been made for the slightly lower calorific value of the boiler fuels as compared with Diesel fuel.

The author's approach was, first of all, to remove from boiler fuel the objectionable matter not found in Diesel fuel and, secondly, to adjust the engine to burn fuel which differed from Diesel fuel only in respect of viscosity. His interest in this problem became greater when, seven years ago, he was made responsible for the technical operation of a very large fleet of tankers, the majority of which were propelled by Werkspoor-type four-stroke engines of 4000 Ihp. These ships, like all other motor ships, were using Diesel fuel, and the opportunity to put the author's theories to the test came three years ago, when the directors of the Royal Dutch Shell Group gave their consent and their money to a full-scale attempt to solve this age-old problem.

Instead of constructing a miniature of the type engine in which the author was chiefly interested and carrying out the experiments on shore, or attempting to experiment with a ship at sea, both of which methods had obvious disadvantages, an exact duplicate of one unit was constructed and installed at the St. Peter's works of Messrs. R. and W. Hawthorn, Leslie and Co., Ltd., who provided, at their own cost, a building to house the engine, and whose keen co-operation and helpfulness are here acknowledged.

The only really objectionable constituent in boiler fuels when used for Diesel engines is ash, which is composed of silica, iron oxide, vanadium oxide and other abrasive matter, and if this ash could be extracted from the fuel before it is injected into the engine cylinders there is every reason to hope that distilled fuels will no longer be necessary for Diesel engines.

Whilst the experimental engine was under construction and the foregoing tests were being carried out, a start was made to see what could be done towards extracting the ash from various grades of boiler fuel sent to the works of Messrs. Alfa-Laval and Co., Ltd., who placed their laboratory and their wide knowledge of centrifuging at the author's disposal. A series of tests were carried out, and whilst these were not conclusive owing to the particular size of centrifugal machines which were proved to be necessary not being available, they showed that whilst these machines were extremely effective in separating water and suspended matter, even of colloidal form, the desired high degree of extraction could not be obtained by centrifuging at usual temperatures and in the customary manner.

The main points brought out during these fuel-purifying experiments were that:—

(1) To obtain the greatest extraction of ash the fuel must be heated to the highest practicable temperature before treatment and maintained at that temperature during treatment.

(2) The fuel should not be kept at this high temperature longer than necessary, as it had the effect of driving off some of the lighter constituents and unduly raising the viscosity of the treated fuel.

(3) If the temperatures were too high certain asphaltic constituents in the fuel, capable of being burnt satisfactorily, tended to separate.

(4) The larger the diameter of the centrifugal separator bowl, the greater was the amount of ash extracted from the fuel, the amount extracted being proportional to the length of time the fuel remained in the bowl.

In regard to (4), this condition was met by employing standard De Laval centrifugal separators and reducing the throughput to half their rated capacity. The machines finally employed had a rated capacity of 700 gallons per hr., but the throughput was regulated to 350 gallons per hr.

Upon reviewing the results obtained during these tests, it was decided that, in addition to purifying in the ordinary way, to remove all water and suspended matter a machine capable of greater extraction must be provided, and that this should take the form of an additional centrifugal separator with a bowl of special construction. Preliminary tests with this machine proved that it was capable of extracting matter which the ordinary form of centrifugal separator was unable to extract, but the bowl construction was such that it was necessary first to run the fuel through an ordinary centrifugal separator at a certain temperature and rate. The ordinary machine is called the "purifier" and the machine with a special bowl the "clarifier."

The clarifier developed by Alfa-Laval for the removal of the last of the ash in the fuel is outwardly similar to the standard centrifugal purifier. The differences between the purifier and clarifier bowls are shown in Figure 1,

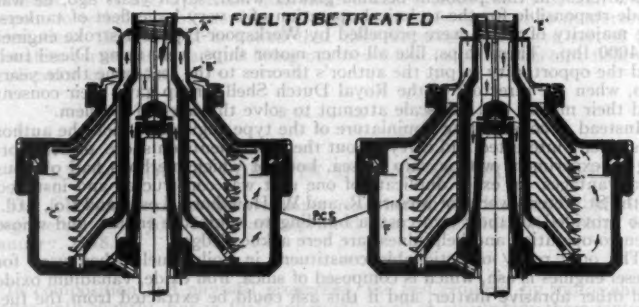


Fig. 1.—Purifier bowl (left). Clarifier bowl (right).

from which it will be observed that, whilst the purifier bowl has two outlets, "A" for purified fuel and "B" for separated water and solid matter, the clarifier bowl has only one outlet, "E," and this is for clarified fuel. The clarifier bowl is not intended to deal with water, and the solid matter separated from the fuel is retained in the dirt space "F" and must be removed periodically by hand. The purifier, therefore, removes the water and particles of earthy matter from the fuel, and the duty of the clarifier is to extract the extremely fine particles and some of the slow-burning constituents from the fuel.

All the discs "PC5" in the purifier bowl are provided with holes near the periphery and separation takes place at these holes. The water and solid matter are thrown outward into the dirt space "C," and the fuel travels upwards, as indicated by the arrow heads. In the bottom disc of the clarifier bowl the holes are omitted, so that those provided in all other discs are screened and the fuel to be treated must flow to the outer edge of the discs before it can travel upwards and inwards. The fuel is thus carried a greater distance from the center of the bowl and a proportionately higher degree of centrifugal force is applied. As the clarifier is not required to separate water from the fuel, the upper end of the bowl is somewhat different from the corresponding end of the purifier bowl. A water seal is necessary in the purifier, but not in the clarifier.

Before beginning the purifying process, the temperature of the fuel is raised to 180 degrees F., which has the effect of reducing the viscosity of 1500 secs. Red. I at 100 degrees F. fuel to 155 secs. As already mentioned, the initial temperature is important as too low a temperature makes separation of water and fine earthy matter more difficult, whilst if too high the very slow-burning asphaltic contents of the fuel which should be extracted becomes so fluid that they would pass through the machines with the fuel supplied to the engine. To obtain the highest degree of purification the fuel should, therefore, be maintained at 180 degrees F. from leaving the storage tank until it reaches the tank from which it flows to the engine. A matter of 10 degrees F. higher or lower will not have a serious adverse effect, but such a range should be considered the permissible limit.

An examination of the material collected in the purifier and clarifier bowls during the final run in one series produced the following information regarding its composition:—

Material.	From Purifier. Per cent.	From Clarifier. Per cent.
Water	35.0	47.8
Carbonaceous matter	16.16	15.47
Ash	48.84	36.73

The ash constituent of the two samples was analyzed and its composition found to be:—

Material.	From Purifier. Per cent.	From Clarifier. Per cent.
Silica	50.65	19.83
Barium oxide	1.33	—
Lead oxide	1.80	0.52
Copper oxide	0.14	0.09
Iron oxide	18.84	10.15
Aluminum oxide	14.38	7.84
Titanium oxide	0.42	0.42
Vanadium oxide	—	0.06
Calcium oxide	1.69	1.28
Magnesium oxide	0.13	0.73
Sodium oxide	7.20	0.90
Sulphur (as SO_3)	3.13	9.57
Phosphorus (as P_2O_5)	0.27	0.28
Sodium chloride	—	41.86
Undetermined	0.02	6.47
	100.00	100.00

The ship selected for the second stage of this interesting experiment was the 12,250 tons deadweight 12-knot single-screw tanker *Auricula*. The principal particulars of the airless-injection supercharged four-cycle Hawthorn/-Werkspoor main engine installed in this ship are:—

Number of cylinders	8
Diameter of cylinders, mm	650
Stroke of pistons, mm	1400
Rpm., rated	115
Ihp., rated	4000
Cylinder compression pressure, lb./sq. in.	490
Cylinder maximum pressure, lb./sq. in.	610
Supercharge air pressure, lb./sq. in.	5.5

Other information regarding this engine which has a bearing on the experiments is that the cylinders, pistons, exhaust valves and fuel valves are cooled by fresh water, and that the fuel system incorporates a surcharge pump, which draws the fuel from the daily service tank and discharges it at a pressure of about 25 lb./sq. in. to the main engine Hp. fuel-pump suctions.

The engine is of standard design and was constructed to operate on Diesel fuel. A large number of these engines have been, and are being, installed in exactly similar ships, and the practice is to run the engines in service at 112-113 Rpm., which corresponds to a power output of 3700 Ihp. During fitting-out the fuel-purifying plant was installed and the fuel system modified to permit the engine being started up on Diesel fuel and changed over to heavy fuel without varying the speed after the engine cylinders and pistons had attained normal working temperatures. A pictorial view of the purifying plant and the fuel system is shown in Figure 2, which, it will be assumed, is sufficiently clear to be self-explanatory.

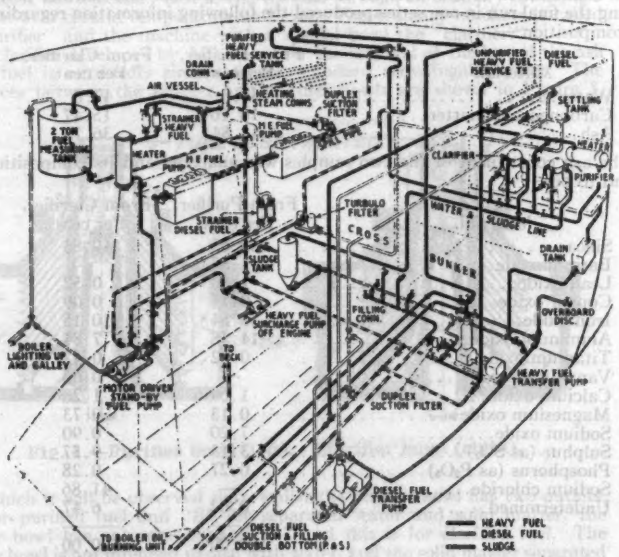


Fig. 2.—Main engine fuel system in the "Auricula."

The *Auricula* underwent sea trials on August 12, 13 and 14, 1946, and began the maiden voyage to Curacao, Dutch West Indies, three days later. The first day of the sea trials was occupied in tuning-up on Diesel fuel.

During the first and second days of the sea trials the averages of two six-hour tests, one on Diesel fuel and one on heavy fuel, are shown in Table 2.

Table 2.

	Diesel Fuel	Heavy Fuel
Specific Gravity of fuel at 60°F	0·86	0·95
Viscosity Red. I at 100°F.	43 secs.	1,225 secs.
Cylinder compression pressure	531lb.	532lb.
Cylinder maximum pressure	633lb.	605lb.
M i.p	109·7lb.	115·5lb.
R.p.m.	120	122
Fuel lever position	14	14
I.h.p.	3,770	3,900
Exhaust gas temperature	308°C.	318°C.
Supercharge air pressure	5·4lb	5·4lb.
J.C.W. pressure	19lb.	20lb.
P.C.W. pressure	22lb.	22lb.
J.C.W. outlet temperature	130°F.	130°F.
P.C.W outlet temperature	119°F.	119°F

No alteration of any kind was made after the first test, the power developed on the second test being governed by the position of the fuel lever, which was in the same position as during the first test. The ship at the time of each test was only about half-loaded, which accounts for the comparatively high Rpm. in relation to power developed.

The greater power developed when on heavy fuel with unaltered fuel-lever position is accounted for by the higher specific gravity, resulting in a greater weight of fuel being injected. The lower maximum cylinder pressure on heavy fuel indicates that the burning process is less rapid than with Diesel fuel, as was found to be the case with the experimental engine.

The bunkers shipped at Curacao for the return voyage had the characteristics shown in Table 3. The only stipulation made was that the viscosity should not exceed 1500 secs. Redwood I at 100 degrees F.

Table 3.

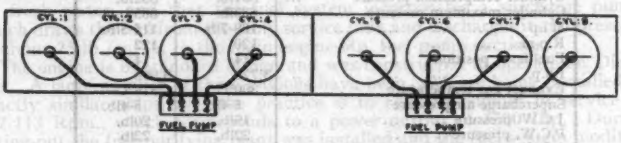
	Outward	Homeward
Specific Gravity at 60°F.	·954	·973
Viscosity Redwood I at 100°F	1,223	1,220
Flash Point Closed °F.	160	185
Carbon Residue Conradson % weight	9·35	8·7
Sulphur	2·00	2·45
Asphaltenes	4·70	—
Ash	0·07	0·04
Sediment by extraction	0·02	0·06
Water % volume	0·50	0·25

During the first six days of the homeward voyage the engine operated as on the outward voyage, the daily averages for this period being as follows:—

Hours engine operating	24
Hours fuel-purifying plant working	14
Rpm.	118.2
Fuel-injection pressure average	8300 lb.
Maximum cylinder pressure average	650 lb.
Exhaust-gas temperature average	338 deg. C.
Color of exhaust gases	Clear
Position of fuel lever	15
Supercharge air pressure	5.7 lb.
Speed of ship	12.5 knots

Subsequent tests carried out on the *Auricula* revealed that the length of pipe between the fuel pump and the fuel valve had a surprisingly great effect upon the injection pressure being shown in Figure 3. The pressures shown are the averages of readings taken daily over a period of three months, and although the pressure indicator used was correct only to within plus or minus 500 lb., the ratios between the cylinders are correct.

The *Auricula* began the outward leg of No. 3 voyage on November 21, 1946, burning heavy fuel having the properties shown in Table 4.



FUEL PUMP	Nº1	Nº2	Nº3	Nº4	Nº5	Nº6	Nº7	Nº8
LENGTH PIPE, FT.	11	7·5	5·5	7	7	5·5	7·5	11
PRESSURES lbs/in²	6200	5600	5400	5900	5900	5400	5700	6100

Fig. 3.—Effect on fuel injection pressures of varying lengths of pipe.

Table 4.

	Outward	Homeward
Specific Gravity at 60°F.	·963	·975
Viscosity Red. I at 100°F. secs. ...	1,204	1,472
Flash Point Closed °F.	212	176
Carbon Residue Conradson % weight	8·8	10·8
Sulphur Content	2·49	2·59
Ash	0·03	0·05
Sediment by extraction...	0·03	0·05
Water ... ... % volume	0·05	0·02

During the first half of this voyage across the Atlantic exceptionally heavy weather was encountered, the average propeller slip for the first seven days being over 25 per cent, and although the Rpm. varied between 70 and 130 because of the pitching motion of the ship, the engine operated as it would have done on Diesel fuel. During the second half of the voyage much fog, necessitating periods of slow running, was encountered, and on these occasions the engine was operated on Diesel fuel, as so far the opportunity to carry out the full prearranged slow-running tests had not presented itself. The voyage took 449 hrs., during 405 of which the engine operated on heavy fuel.

Upon arrival at Curacao Nos. 6 and 8 fuel valves were removed for examination. These valves had been in use throughout the voyage and it was interesting to observe that, unlike the valves removed at sea after the eight hours' slow-running test, the nozzles were quite free of deposit. It is reasonable to assume that these two valves were in much the same condition as regards deposit as Nos. 1 and 3 after the eight hours' slow-running period. If this assumption is correct, then this test proved two things, the first being that combustion of the purified fuel was so good at full power that there was no tendency for the deposit to accumulate and, secondly, that the effect of working up to full speed immediately afterwards was to burn off the deposit which had accumulated during the slow-running period. The practice at that

time of changing over to Diesel fuel for maneuvering into port, and the possibility of this grade of fuel being responsible for burning off the deposit, was not overlooked.

Alterations were made to the supercharge air system also with the object of enabling the engine to operate on supercharge air pressure varying between 2½ and 5½ lb./sq. in. without restricting the flow of air to the pumps.

Whilst the ship was discharging cargo the alterations mentioned were carried out, some being of a temporary nature in order to avoid expense and delay. For instance, experience suggested that the fuel-valve cooling system should be entirely separate (Figure 4) instead of being connected to the piston-

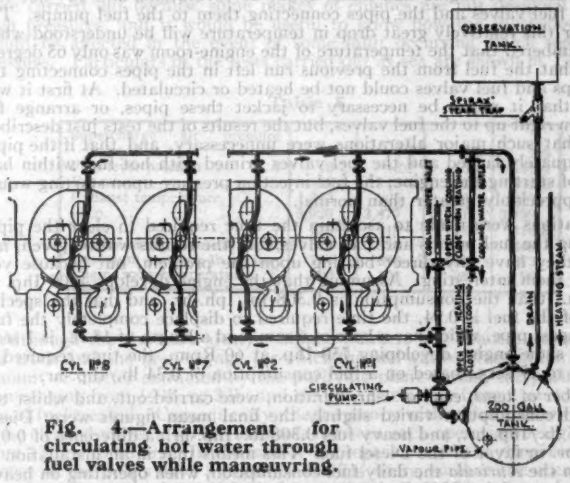


Fig. 4.—Arrangement for circulating hot water through fuel valves while manœuvring.

cooling system in order to reduce the time required to preheat the water and enable the temperature of the cooling water to the fuel valves to be raised to a temperature higher than that advisable for the pistons. Instead of separating the systems, the arrangement provided only for heating the whole of the piston-cooling water by steam injection.

Preheating Cylinder Jackets.

When all work had been completed and the ship adequately moored, the steam jets raised the temperature of the cylinder-jacket cooling water to 120 degrees F. in 6 hrs. and the combined piston and fuel-valve cooling water to the same temperature in 3 hrs., the initial temperature of the water in each case being 60 degrees F. The heavy fuel in the service tank was raised to a temperature of 160 degrees F. and circulated through the new heater, which had been installed between the service tank and the engine, until the temperature at the main engine fuel-pump suction was 180 degrees F. Circulating in this way is possible in the Werkspoor engine, as in most others, by spilling to the service tank.

Although the temperature of the engine-room was not more than 65 degrees F., the engine started up and began working regularly on heavy fuel with the expenditure of no more starting air than would have been required with Diesel fuel. To approximate service conditions as nearly as possible, and to obtain the data required, the speed of the engine was not allowed to exceed

55 Rpm., and this speed was maintained for 30 mins., during which time every cylinder "fired" with the utmost regularity and, to all outward appearances, would have continued doing so indefinitely. The temperature of the fuel at the engine fuel pumps was 175 degrees F. throughout this test, and the estimated drop in temperature before reaching Nos. 1, 4, 5 and 8 fuel valves, which are situated farthest from the fuel pumps, was 10 degrees F.

No provision was made to register accurately the temperature of the fuel as it entered the fuel valves, but the estimated temperature on first starting after 1½ hrs. at rest was 100 degrees F. This low temperature and correspondingly high viscosity was evident from the heavier pumping action, which occurred until the engine had made about 20 revolutions and the hot fuel filled the fuel valves and the pipes connecting them to the fuel pumps. The reason for this relatively great drop in temperature will be understood when it is remembered that the temperature of the engine-room was only 65 degrees F., and that the fuel from the previous run left in the pipes connecting the fuel pumps and fuel valves could not be heated or circulated. At first it was thought that it might be necessary to jacket these pipes, or arrange for circulation right up to the fuel valves, but the results of the tests just described proved that such major alterations were unnecessary, and that if the pipes were adequately lagged and the fuel valves primed with hot fuel within half an hour of starting the engine, the fuel injection pressure upon starting would not be appreciably greater than normal.

Calculations were made to ascertain the time required to clear the pipes connecting the fuel pumps and fuel valves, and the results will be given, not because they have any direct bearing upon the problem, but because you may find them interesting. Assuming that the engine develops 4000 Ihp. at 114 Rpm., that the consumption is 0.308 lb./Iph.-hr. and that the specific gravity of the fuel is 0.94, the time required to displace completely the fuel in the longest pipe, which has a bore of 6 mm. and a length of 15 ft., is 7 secs. With the same engine developing 750 Ihp. at 60 Rpm., the time required is increased to 17 secs., based on a fuel consumption of 0.34 lb./Ihp.-hr.

A number of tests, each of 1-hr. duration, were carried out, and whilst the calculated consumption varied slightly the final mean figures were: Diesel fuel 0.295 lb./Ihp.-hr., and heavy fuel 0.308 lb./Ihp.-hr., a difference of 0.013 lb./Ihp.-hr. in favor of the Diesel fuel. This means that in an installation of the size in the *Auricula* the daily fuel consumption, when operating on heavy fuel, is increased by 1248 lb. (0.56 ton) per day. Part of this increase is accounted for by the difference in calorific value, which, for the tests referred to, was 18,950 Btu.'s per lb. for Diesel fuel and 18,600 Btu.s per lb. for heavy fuel. After making allowance for this difference, it will be found that, in the case of the *Auricula*, the daily consumption of heavy fuel is 704 lb. (0.31 ton) greater than the daily consumption of Diesel fuel.

No attempt has been made to ascertain the quantity of fuel represented by the increase of 12 degrees C. in the temperature of the exhaust gases, which occurs when changing over from Diesel fuel to heavy fuel, but there is no doubt that much of the 13,100,000 heat units in 704 lb. of fuel will be found in the exhaust gases, in which case they serve a useful purpose if, as is common practice, the gases pass through a boiler for the generation of steam.

At prices given in Table 1 for Diesel fuel and No. 2 heavy fuel, i.e., 83s. and 61s. per ton respectively, the daily consumption of heavy fuel of a 4000 Ihp. Diesel engine could be 4.56 tons greater than that of Diesel fuel before there would be no advantage—all other considerations being equal—in changing over a ship to heavy fuel. In other words, if the consumption of Diesel fuel is 12.6 tons per day, the consumption of No. 2 heavy fuel would require to be 17.16 tons per day before the fuel bills would be equal. Actually, the consumption of heavy fuel in such an installation would be 13.16 tons per day, so that the saving amounts to 4 tons per day, which in cost is over £200 on each Atlantic crossing. In an 8000 Ihp. installation this saving would, of course, be doubled.

In common with all Werkspoor-type four-cycle engines, the engines of the *Auricula* are provided with the under-piston supercharge system, which is so simple in construction, reliable in service and economical in upkeep. The normal supercharge air pressure at full speed is 5½ lb./sq. in., and an adequate amount of air for efficient operation is automatically provided at all speeds, so that in order to operate at full speed with supercharge pressures below the normal figure it was necessary to arrange controls which would return a proportion of the discharged air to the suctions of the supercharge air pumps. To have throttled the pump suctions would have affected the mechanical efficiency of the engine, besides tending to draw lubricating oil out of the crankcase. Tests with a lower supercharge air pressure than 5½ lb./sq. in. were carried out because some makes of engines are not designed for such high pressures, and it would be useful to know how the Werkspoor engine would be affected in the event of some fault occurring which reduced the pressure. Particulars of the tests carried out at sea are shown in Table 5.

Table 5.

Supercharge air pressure lb./sq.in.	5·5	4·5	3·5	2·5
Fuel lever position	13·2	13·2	13·2	13·2
R.p.m.....	113·5	113·4	112·6	112·6
Exhaust temperature, C ..	312	325	340	357
I.h.p.	3,920	3,877	3,791	3,759
M.i.p.	120·6	119·4	117·6	116·6
Compression pressure lb. sq.in.	543	521	494	469
Maximum pressure lb. sq.in.	714	698	685	673
Engine room temperatures, °F	100	101	98	96

Although the *Auricula* always shipped bunkers at Curacao, except for the sea trials and the first voyage, for which the fuel was received in the U.K., the amount of matter extracted by the purifying plant varied, the purifier's portion ranging from ½ lb. to 2 lb. and that of the clarifier from 2 oz. to 5 oz. per ton of fuel treated. Consequently, the frequency with which the bowls require to be cleaned varies also, the intervals between each operation being anything from 12 hrs. to 48 hrs. In the case of the separators on the *Auricula*, the dirt spaces of the bowls have a capacity of 10½ liters, so that the accumulation would require to be about 20 lb. before separation would be adversely affected.

On the seventh voyage of the *Auricula* the bunkers shipped at Curacao made it necessary to clean both the purifier and clarifier bowls every 2 hrs., the total amount of matter extracted averaging 25 lb. per ton of fuel treated. Although it was obvious that the fuel was "breaking down" under the treatment, the ship was allowed to proceed as the engines were operating perfectly on the purified fuel. The thorough investigation which followed revealed that this particular shipment was derived from mixed paraffin base crudes and asphaltic base crudes, and that whilst such a mixture was stable at normal atmospheric temperatures, "breakdown" occurred at 175 degrees F. It is most unusual refinery practice to mix such crudes and, the author is informed, quite unnecessary.

During the voyage, when the need to clean the bowls became much too frequent it was found that running hot fresh water through for about 5 mins., in place of the fuel, softened the deposit in the purifier bowl and caused much of it to pass out of the sludge outlet. Illuminating oil, introduced in the same way, produced even better results. With a normal grade of 1500 secs. fuel and the introduction of 1-2 gallons of illuminating oil at about 12-hr. intervals, the bowls will run for several days before requiring to be opened up. The best results will be obtained if the illuminating oil is introduced very slowly. The same applies, of course, if hot water is used. The construction of the clarifier bowl does not permit such treatment, but as the quantities of material separated by this machine are so small, compared with the quantities extracted

by the purifier, the clarifier will operate at full efficiency for days, probably a week, before opening up for cleaning becomes necessary.

Excepting for the voyage when unstable fuel was shipped, the practice on the *Auricula* was to open up the bowls once daily, and the time taken to carry out the operation was, roughly, half an hour. Much can be done to make this work less unpleasant and reduce the time to carry it out if, during installation of the plant, due regard is given to the size and position, relative to the machines, of the bench and receptacles for the extracted material, as well as arranging for all necessary tools to be within reach when in their rightful positions. A further advantage is to have the purifier and clarifier so arranged that it is possible for the operators to work at them from all sides. In view of the close proximity of fuel heaters, etc., the space occupied by the purifying plant should be well ventilated.

The usual practice in marine Diesel engines is for the fuel to flow by gravity from the service tanks to the main engine Hp. fuel pumps. In some cases a surcharge pump is provided, as on the Werkspoor-type engines. With Diesel fuel, and where a reasonable pressure head can be produced at the fuel-pump suctions, mechanical means of creating the requisite pressure is unnecessary, but with heavy fuel such means may be advisable, particularly if the distance between the service tank and the fuel pumps is great and/or indirect. On the *Auricula* it was found that a pressure of 25 lb./sq. in. at the pump suctions ensured 1500 secs. fuel following up the pump plunger satisfactorily at full speed. A test carried out with a pressure of 45 lb. at the pump suctions caused the engine revolutions to be increased 1.9 Rpm., which indicates "slack" oil at the pump suction with a pressure of 25 lb., but the engine operated with perfect regularity at the lower pressure. A higher pressure is not advocated unless the desired results cannot be obtained.

As in the case of all other parts, the design of the main engine Hp. fuel pumps of the *Auricula* was in accordance with standard practice. During the early stages of the experiment the spring load on the plungers was increased to overcome a tendency to stick, due to the film of heavy oil adhering to the rubbing surfaces being thicker than would be a film of Diesel fuel. Later, the spring load was reduced to normal, as it was found that sticking could be prevented by making the plungers a slightly easier fit. The tolerance which was found to obviate sticking, while at the same time preventing leakage when the heavy fuel was heated to 175 degrees F., was between 0.00015 in. and 0.00020 in. The finish of the contact surfaces of plungers and bushes must be perfectly smooth and free from "holidays," otherwise certain resinous substances from the hot fuel will deposit and may eventually cause the plungers to stick.

Cost of Equipment.

Turning now to the cost to equip a ship for the burning of heavy oil, the *Auricula* had all her machinery installed and was within a month of completion when it was decided to convert that part of the installation concerned. Moreover, this ship was provided with experimental gear and equipment which would not be required in subsequent conversions. Even so, the total extra cost to fit out the *Auricula* was £7259, which includes the cost of the two centrifugal separators, heating coils in storage and service tanks, etc. So far no existing ships have been converted, owing solely to the difficulty of obtaining the electrical portion of the separators, but several new ships have been fitted out, and even at present-day inflated prices the cost is under £4000. The cost to convert existing ships will vary according to the arrangement of fuel storage and service tanks, but if the arrangement follows general practice it is estimated that the cost will not exceed £5000, which is approximately the saving effected during one year's operation of a 4000 Ihp. installation. The cost to convert will not be materially affected by the power of the installation, so that for an 8000 Ihp. installation the initial outlay will be recovered after six months' service.

The equipment for a 4000 Ihp. installation comprises the following:—

1. Steam-heating coils in storage and service tanks capable of maintaining fuel at 120 degrees F.
2. One purifier with self-driven suction and delivery pumps. One clarifier with self-driven delivery pump only.
3. One multi-tubular heater, located between service tanks and purifier, capable of heating fuel from 100 degrees F. to 180 degrees F. at 200 gallons an hr.
4. One multi-tubular heater, located between service tank and Hp. fuel pumps and of similar capacity to item 3.
5. One effective strainer located between heater, item 4 and Hp. fuel pumps.
6. One work bench and tool rack, situated close to separating machines with two 10-gallon receptacles, one for cleaning oil and one for sludge.
7. One receptacle for water and sludge discharged by purifier whilst operating.
8. Hot-water circulating system for fuel valves.

Between August 17, 1946, and August 13, 1947, the *Auricula* accomplished the following:—

Distance covered between pilot vessels.....68,715 miles
Hours main engine operated at full speed.....6,136 hrs.
Total revolutions made by main engine.....42.33 million

At the end of this period Nos. 3, 4, 5 and 6 cylinders were opened up and the cylinder liners, piston rings and piston-ring grooves gauged for wear, but before giving figures the author would mention again that the second voyage (8483 miles, 722 hrs. 5,000,000 revolutions) was made on Diesel fuel. Also that the cylinder liners, piston rings and piston-ring grooves were not, unfortunately, microscopically gauged during the construction of this engine. The question of opening up and obtaining accurate measurements of these parts after the sea trials was considered, but owing to the urgent need for the ship's services the seven to eight days' delay that would be incurred could not be allowed, so that when calculating the wear rates now to be given it had to be assumed that the parts concerned were machined to the exact dimensions given on the working drawings.

The upper ends of all cylinder liners were entirely free of deposit, as was the under side of the cylinder head, excepting for a patch of deposit between the fuel-valve and exhaust-valve pockets. This deposit was obviously the result of running the engine at slow speed and the fuel valves at too low a temperature, reference to which was made earlier in this paper.

With the exception of No. 4, the bores of all cylinder liners examined on this occasion were perfectly smooth and polished, and it was apparent that no gritty matter had been present in the cylinders and that the normal grade and quantity of lubricating oil used was efficient and adequate. Moreover, the surfaces were more greasy than is generally the case with Diesel fuel and the appearance raised the hope that the use of heavy fuel had assisted lubrication of the cylinders, and that consequently the amount of wear would be found to be very small. The oil used was Shell C3, and the quantity ¾ gallon per cylinder per day. The surface of No. 4 liner was equally smooth, but dull and dry, due to water having accidentally entered this cylinder whilst the head valves were being removed.

The original diameter of these liners was 650 mm., and the construction is shown in Figure 5. This illustration also indicates the points at which the wear was gauged, while the gauge readings after one year's operation are shown in Table 6.

From the foregoing gaugings it will be seen that the mean wear at the point A, where the greatest wear usually takes place, is:—

No. 3 cylinder.....0.665 mm.
No. 4 cylinder.....0.664 mm.
No. 5 cylinder.....0.770 mm.
No. 6 cylinder.....0.495 mm.

The mean wear of these four cylinder liners is 0.648 mm., and as the ship had covered 68,715 miles the engine operated for 6136 hrs. and made 42.33

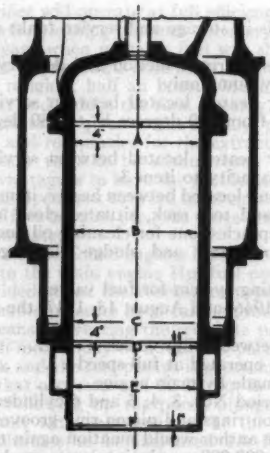


Fig. 5.—Points gauged for liner wear in the "Auricula."

Table 6.

	Cylinder No. 3		Cylinder No. 4		Cylinder No. 5		Cylinder No. 6	
	F. & A.	P. & S.	F. & A.	P. & S.	F. & A.	P. & S.	F. & A.	P. & S.
A	650·52	650·73	650·70	650·63	650·61	650·93	650·45	650·54
B	·42	·75	·31	·53	·47	·82	·32	·50
C	·57	·80	·45	·49	·57	·87	·42	·51
D	·69	·69	·49	·53	·61	·93	·45	·54
E	·69	·75	·46	·61	·66	·76	·23	·43
F	·35	·52	·30	·34	·41	·43	·14	·14

million revolutions, the wear rates per 1000 miles, per 1000 hrs. and per 1,000,000 revolutions being as follows:—

Per 1000 miles.....0.0093 mm. or 0.0036 in.
Per 1000 hrs.....0.1040 mm. or 0.00409 in.
Per 1,000,000 revs.....0.0150 mm. or 0.00059 in.

It goes without saying that the wear rates of cylinder liners and piston rings depend as much, if not more, upon the materials of which these parts are made as upon the proper burning of the fuel, so that if the difference due solely to running on boiler fuel is to be ascertained the wear rates just given should be compared with the four cylinders of an exactly similar engine which had operated for the same length of time on Diesel fuel.

The *Goldmouth* is propelled by an exactly similar engine, and when Nos. 2 and 7 cylinders were opened up and the liners gauged for wear, the engine had operated 5161 hrs. at full speed and made 33,549,300 revolutions on high-grade Diesel fuel. The average wear of these two liners at the point indicated by A in Figure 20 was 0.505 mm. and the wear rates are :—

Per 1000 hrs.....0.0980 mm. or 0.00386 in.
Per 1,000,000 revs.....0.0150 mm. or 0.00059 in.

It is usual to speak of wear rates as so much per 1000 hrs., but as this basis varies according to the rotational speed of the engine and to the nature of the voyage, the steaming time being calculated between pilot vessels, the wear

rate per 1,000,000 revolutions gives a more accurate comparison and, as will be observed from the figures just given, the wear rates of the *Auricula* cylinder liners are exactly the same as those of the *Goldmouth*, notwithstanding the former ship having operated on boiler fuel and at 5 per cent greater power output.

Assuming that the average revolutions made by an engine-burning boiler fuel is 40,000,000 per annum, and that the maximum allowable cylinder-liner wear is 6 mm., the engine will operate over 10 years before renewal of liners becomes necessary when conditions are similar to those prevailing on the *Auricula*. These conditions were of necessity far from ideal, since apart from running the engine for quite long periods with reduced fuel-injection pressure, low fuel temperatures, etc., the weather encountered for eight months out of the 12 was the worst possible. Furthermore, there is every reason to believe that the wear-resisting qualities of the piston rings employed were not of the best.

Nos. 3, 4, 5 and 6 pistons were disconnected from the piston rods and every part carefully examined. The nuts of Nos. 3 and 5 pistons were found to have slackened slightly, and it was evident that water had got between the joining surfaces. Water could only emanate from leaking piston-cooling telescopic-pipe glands, and water from this source would also find its way on to the cylinder liners. Moreover, it was evident that these two pistons had been rubbing lightly against the liners, so that how much these two faults were responsible for liner wear can only be surmised.

All four pistons were entirely free from deposit of any kind and their remarkably clean condition can best be illustrated by the statement that, after wiping with a paraffin-soaked rag, they were ready to be refitted.

Every piston ring (24 in number) in each of the four pistons examined was free to the extent that, in addition to being easily moved in and out of the grooves, they could be moved circumferentially in their grooves with ease. The original width of the piston-ring grooves, as given on the working drawing, was 14.05 mm. for the top groove and 14 mm. for all the others, the corresponding dimensions of the piston rings being the same, less an allowance for working clearance, and when gauged after a year's operation the widths of the grooves were found to be as shown in Table 7.

Table 7.

Grooves	Top	2nd	3rd	4th	5th	Bottom
No. 3 Piston ..	14·15	14·15	14·15	14·10	14·10	14·15
No. 4 ,, ...	·15	·10	·10	·10	·10	·10
No. 5 ,, ...	·15	·15	·10	·10	·15	·10
No. 6 ,, ...	·20	·15	·15	·10	·10	·15

The original radial thickness of the five upper piston rings was 21 mm., and when those in Nos. 3, 4, 5 and 6 pistons were gauged after one year's operation the radial thicknesses were found to be as shown in Table 8.

Each piston ring was gauged at nine points and the greatest wear was found to have taken place directly opposite the gap. The opening between the ends (gap) of each ring when inserted in an unused cylinder liner is shown in Table 8, the original opening being 2 mm. The wear of the sixth ring is not given in Table 8, as this is the oil-scraper ring and is chamfered for half its depth.

It will be observed that the wear of each of these concentric rings is very uneven, the difference being nearly 4 mm. in some rings. In every case the greatest wear occurred at a point opposite the gap and the least at each side of the gap. This proves the earlier statement that all piston rings were found quite free in their grooves, and that the cause of this uneven wear is due entirely to the uneven outward pressure of the rings.

Table 8.

	Radial Thicknesses			Mean Wear	Gap
	Max.	Min.	Mean		
				mm	mm
No. 3 Piston					
Top Ring ...	16·2	14·2	15·20	5·80	35·5
2nd ,, ...	17·6	14·3	15·95	5·05	33·0
3rd ,, ...	17·7	15·6	16·65	4·35	26·0
4th ,, ...	18·7	16·6	17·65	3·35	22·0
5th ,, ...	19·1	17·9	18·50	2·50	16·5
No. 4 Piston					
Top Ring ...	17·3	14·7	16·00	5·00	32·0
2nd ,, ...	18·0	15·0	16·50	4·50	27·5
3rd ,, ...	18·6	14·7	16·65	4·35	25·5
4th ,, ...	18·6	16·5	17·55	3·45	20·0
5th ,, ...	18·8	17·1	17·95	3·05	18·5
No. 5 Piston					
Top Ring ...	16·6	14·3	15·45	5·55	35·0
2nd ,, ...	17·9	15·5	16·70	4·30	27·0
3rd ,, ...	18·1	16·1	17·10	3·90	25·5
4th ,, ...	18·6	17·0	17·80	3·20	21·5
5th ,, ...	18·8	17·5	18·15	2·85	19·0
No. 6 Piston					
Top Ring ...	18·2	15·1	16·65	4·35	27·5
2nd ,, ...	18·2	16·5	17·35	3·65	23·0
3rd ,, ...	18·1	16·4	17·25	3·75	21·5
4th ,, ...	19·0	17·1	18·05	2·95	19·0
5th ,, ...	18·9	17·6	18·25	2·75	17·0

When the exhaust valves were removed they had been in continuous use for the following full-speed running periods, the maneuvering time being additional:—

Valve No.	Hrs.
1	2022
2	1400
3	1400
4	2516
5	560
6	2516
7	4768
8	2022

All valve housings were in the condition that would be expected had the engines been operating on Diesel fuel, the only deposit found being on the outside at the lower end and extending for about 2 ins. above the landing. All spindles were an easy fit, and after the securing nut had been removed were easily withdrawn by hand.

No deposit of any kind was found in the exhaust-gas manifold, the internal surface of which was evenly covered with a very thin film of dry soot which could be removed with the fingers.

When the eight fuel valves were removed they had been in continuous use for the following full-speed periods:—

Valve No.	Hrs.
1	990
2	2678
3	1596
4	560
5	560
6	1380
7	4768
8	990

After removal, No. 2 fuel valve, which had been in use for 2678 hrs., was connected to the test pump and tested for fluid-tightness and spraying with Diesel fuel. This test proved the needle valve to be perfectly fluid-tight and not only all spray holes clear but that the sprays had the same form as those from a valve which had never been in use. The lifting pressure of the needle valve was found to be 3700 lb./sq. in., or 150 lb./sq. in. less than the designed pressure.

The whole of these experiments have been carried out with Werkspoor four-cycle-type engines, and the question which will naturally arise is "Will similar results be obtained with other makes of four-cycle engines, or even two-cycle engines?" This question cannot be answered yet, but the author sees no reason why similar results should not be obtained with other makes of four-cycle engines, or even two-cycle engines, providing the undesirable matter in the fuel used is extracted as efficiently as was done on the *Auricula*.

The author claims, and thinks it has been proved, that, so far as composition is concerned, heavy fuel treated in the manner here recommended is no more harmful to the vital parts of a Diesel engine than is Diesel fuel. If this is so, then the only difference between purified heavy fuel and Diesel fuel is in respect of viscosity, in which case it is only engine details that might require to be adjusted or altered. The heavy fuel may not burn so rapidly as Diesel fuel unless the injection system is altered to increase the degree of atomization, but it is suggested that the evidence here put forward proves that the formation of the fuel sprays differs only slightly, if at all, and that the whole of the fuel is burnt in the allotted time. If it were otherwise, would not deposit be found in the cylinders and/or the exhaust-gas passages?

AMERICA'S HIGHEST-POWERED MOTORSHIPS.

The following paper is reprinted from "Motorship" for September, 1947.

Wartime construction of marine diesel-electric propulsion machinery for the U. S. Navy, Coast Guard and Army exceeded the aggregate horsepower of motorships built for the Maritime Commission. Their operation under extreme conditions proved their ability to stand the gaff of gruelling service under the care of rapidly trained operators, and confirmed the sound engineering judgment of the Bureau of Ships, Navy Department. Over 100 vessels, including submarines, exceeded 5000 Bhp. per installation, and fifteen ships exceeded 10,000 Bhp. Yet, despite their remarkable economy and reliability, no contemplated program of the Maritime Commission, or construction plans of American private shipowners, included diesel-electric motorships of power approaching that of the submarine tenders discussed in this article. This position offers food for profound thought, because foreign shipyards are full of large motorships under construction, which will form tomorrow's competitors of the U. S. Merchant Marine.

Seven 18,800-ton motorships were completed for service as submarine mother ships and placed in service by the U. S. Navy before the termination of World War II. The high-powered machinery of these vessels formed an important departure from the engineering practices common with the propulsion of large American merchant ships. Similar hulls and engines would make very desirable and economical passenger-cargo liners or fast freighters. Therefore, shipowners should closely study these naval auxiliary craft and carefully compare them with turbine-driven commercial vessels of their size and power, insofar as the machinery installations are concerned. The Cleveland Diesel Engine Division of the General Motors Corporation was selected to equip these seven noteworthy motorships with a total of well over 100,000 Hp. in main and auxiliary engines. Some of them now have had six years' service, so have been well tested.

The interiors of these ships—officially known as "Submarine Tenders"—differ from merchant craft in that they are chock-full of furnaces, foundries, forges, machine shops, pattern shops, optical plants, etc., plus accommodations for an extra large crew of officers, sailors, engineers and mechanics for the purpose of servicing and repairing submarines. Each ship has a top loaded displacement of nearly 19,000 tons, and a full war strength of about 1100 officers and men, as well as accommodations for about 500 men from submarines when their crews need relaxation and rest. At the present time, the ships are operating with crews of peacetime strength.

One outstanding feature is that, although each vessel has about 15,800 Bhp. in diesel engines for her combined main and auxiliary power, and is capable of a speed of 17 knots, the two main diesel-electric engine rooms occupy a joint length of only 66 feet by the width of the hull. There also is a heating-boiler and evaporator room between the two 33-ft. machinery compartments, which a fast cargo liner probably would not need, as arrangements possibly could be made to utilize the waste exhaust gases at sea, and carry the same to a boiler mounted on a flat over the engines. In the case of Navy ships, this has not been done.

Shipowners who operate modern geared turbine, or turbo-electric ships of over 15,000 total horsepower may express doubts when informed of the very low fuel consumption of diesel-electric-driven submarine tenders. At almost top ship speed, the daily fuel consumption for all purposes is under 58 long tons, or 430 bbl. (one ton of Navy diesel oil equals 7.4 bbl.). Exact figures are given later. Incidentally, the Navy records all oil and water supplies in gallons; and for the purpose of this article, the standard conversion figure of 42 gal. to the barrel has been used.

The remarkable economy and resultant extra carrying capacity which a similarly powered merchant ship would thus be given can be understood, because the most modern and efficient geared-turbine installation of the same aggregate power would burn about 100 tons per diem of Bunker C fuel at full load. This is equivalent to 645 bbl., as a ton of heavy fuel oil equals 6.45 bbl. (A turbo-electric steamship would burn nearly 110 tons of oil.) On a 20-day round voyage, this would mean a difference at sea of 840 tons, or a substantial saving in fuel to the advantage of diesel-electric power. Compared with turbo-electric drive, there would be a fuel saving of approximately 1100 tons, and in neither case does this include the saving in fresh water effected by the motorship, because the auxiliary heating boiler would require a comparatively small amount, as also would the closed cooling system for the diesels. And in warm weather, heating would not be required, except for service water.

Twelve round voyages a year would result in an ability to carry a total of 10,000 to 13,000 tons additional profit-earning cargo, plus the very considerable fuel savings in port. Actually, the greater quantity of fuel per ton of diesel oil referred to, plus the port fuel savings, should almost be sufficient to offset the extra cost per ton, especially if the wise shipowner purchases his diesel fuel by the ton, instead of by the barrel, as do European motorship owners.

One very important technical feature of these naval mother ships, to which the attention of shipowners is drawn, is that the diesel engines are of high speed, turning at 750 Rpm. under full load. Experience with the manufacture and operation of high-powered diesel installations, and the recent advance in metallurgy has been so extensive that today all the reliability of massive slow-speed reciprocating engines is obtainable for protracted voyages. The U. S. Navy and Coast Guard together placed in service some 60,000,000 Hp. in high-speed marine diesel engines during the war in many dozens of different types of craft, or more than the aggregate horsepower of geared-turbine and turbo-electric ships built by the Maritime Commission. About 100 of these vessels, including submarines, exceeded 5000 Hp. each, and 15 exceeded 10,000 Hp. The General Motors Corporation powered more of the higher powered vessels than any other diesel manufacturer.

The smaller and lighter parts of fast-running diesels mean a great saving in time and labor when overhauling or making repairs, and certainly lessen the strain on the operating crew, compared with huge slow-speed units. Shorter stays in port for inspection and repairs reduce demurrage charges, and increase the cargo-revenue-earning period. These factors represent dollars and cents in the overall operating costs of a large ship.

It is an error of information to hold a firm opinion that high-speed diesels cannot run for protracted periods without trouble, or that good operators are not available. During the war, the Navy and the Army trained thousands of excellent diesel operating engineers, many of whom have now returned to private life. Shipping men should not allow themselves to be influenced by the results of the diesel engineering era of 20 years ago if our merchant marine is to remain progressive and hold its own in world competition. According to the Navy Department, the submarine tenders have proved reliable under gruelling service, and their average speed is about 15.4 knots. The Bureau of Ships, Navy Department, deserves every credit for its vigorous and sound engineering policy which enabled these splendid motorships to be built.

Consistent with Navy practice of shifting officers to different duties, the *Proteus* has had several captains and five chief engineers during her career to date. Her present commanding officer is Captain J. A. Jordan, and her engineer officer is Lieut. F. Janise. Captain Robert W. Berry, a former commander and now Navy public information officer at San Francisco, recently stated that this ship's cruising speed was around 16 knots, and that no serious trouble was encountered in the engine room during his command in 1944.

Shipowners should not overlook the fact that in some ways the particular work of a submarine mother ship is more severe on the diesel engines than is the operation of a merchant vessel, because aside from long voyages to and from the Pacific Coast, the stoppings, startings, slow, medium, and fast speeds are most frequent and erratic when submarines are maneuvering. The numerous changes in engine operating temperatures impose greater strains than does steady maintenance of the power load. Sudden heavy loads are also thrown upon the auxiliary diesel units, such as when the electric furnaces, big deck cranes, heavy machine tools, etc., are put into action. In an ordinary land engineering works, changes in load are not noticeable, except miles away at the central power station where the generating units are of great power.

In this article, we are dealing mainly with the *Proteus* as being typical of the other ships, some of which have had the two big cranes aft removed. These seven submarine tenders are named as follows: *Fulton*, *Sperry*, *Proteus*, *Orion*, *Howard W. Gilmore*, *Bushnell* and *Nereus*. Two, inluding the *Proteus*, were built by the Moore Dry Dock Company, Oakland, California, and the others were constructed by the Mare Island and Bremerton Navy Yards. With the exception of the *Sperry*, all have d.c. electric propulsion, the *Sperry* having a.c. equipment. Certain advantages of the a.c. drive have become apparent from experience, and probably will result in any future tenders being equipped with this system.

The welded and partly riveted hull of the *Proteus* has very handsome lines and if a similar hull were fitted with superstructure and interior, as required by a passenger-cargo liner, she would make a fine looking craft. As it is, she is very striking in appearance. Her length is 529 feet, 6 inches and she has a beam of 73 feet, 4 inches. She is a combined hotel, hospital, blacksmith shop, power house and modern engineering plant, completely electrified from stem to stern, with the exception of the heating system, which is steam from two boilers. At one time, when she or a sister ship was off a Pacific beach, she fed close to 4000 men from her galleys and baker shop. Her refrigerating plant provides 900 pounds of ice per day and is operated by three motor-driven five-ton compressors. The electric foundry can handle 500-lb. castings. There also are a sheet metal shop and torpedo shop, as well as a printing shop and photographic department. Many dozens of machine tools are installed in the main machine shop, all electrically driven.

As previously mentioned, there are two 33-foot-long engine rooms. The forward machinery compartment contains four Model 16-248 General Motors-Cleveland Diesel two-cycle engines of 1600 maximum Bhp. each at 750 Rpm., with sixteen 8½-in. by 10½-in. V-arranged cylinders.

On the tenders having d.c. electric propulsion, each of the four main engines in the forward room is coupled to an electric generator of 1125 Kw. and the combined current goes to four motors that are reduction-geared to the starboard 13¼-in. diameter propeller shaft. Each of these 550-volt motors can deliver 1440 Hp. at 750 Rpm., or an aggregate of 5760 Hp. per shaft when under full load. As this is duplicated in the after engine room on the port propeller shaft, there is a total of 11,520 Hp. available at the reduction gear for propulsion. Of course, there can be variations of this exact power, depending upon just what is termed maximum power, and as is indicated in the remarks already made.

For long distance operation, a cruising output of 1250 Bhp. per engine is usually adopted with these ships. This gives adequate sea speed for most duties and means that there is surplus power available for other purposes, when necessary. For instance, engines Nos. 1 or 2 can be used to provide current for an 850-Kw. motor-generator unit, while main engines Nos. 5 and 6 in the after compartment can charge submarine batteries in parallel or series, and also can supply 100 Kw. for lighting in the submarine while their batteries are down. By means of careful checking, the engineers of the *Proteus* have found that the most economical fuel curve for the ship is obtained when 4800 Bhp. is being developed with six main engines running and the twin propellers at 68 Rpm., instead of 130, because only 28 gal. of fuel per shipmile are consumed. This is a remarkable figure and could not be approched by the most efficient turbo-electric-propelled vessel of her power, speed and dimensions.

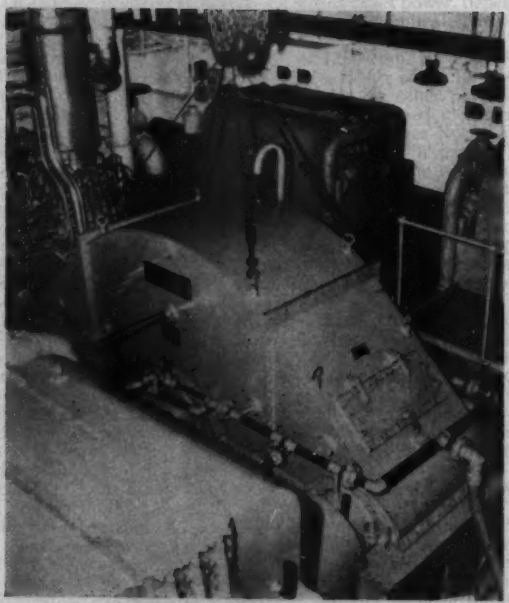
The starboard propeller is of bronze, 15-ft. diameter by 15-ft., 2-in. pitch, and weighs 15,975 lbs. The port propeller and its diesel-electric drive are exactly the same as the starboard units. The length of the starboard shaft, however, is shorter because it starts in the after engine room, whereas the port shaft leads from the forward machinery compartment. Navy practice calls for two complete engine rooms in case of damage by torpedo, bomb or shellfire. Watertight bulkheads and the boiler room separate them completely, and to pass from one engine room to the other it is necessary to go up to the main deck and down again.

Also in the forward engine room are two 12-cylinder Model 12-248 two-cycle, 750-Bhp., 500-Kw., GM-Cleveland Diesel generator sets arranged on the port side, with their control boards on a flat just forward. These are for auxiliary power. The main panel for the propulsion units is on the starboard side of the same flat. Other equipment consists of the large motor-generator set, four air compressors of 30 cu. ft. at 150 lbs., bilge, fire, service, circulating fuel and lube oil pumps, strainers, filters and two centrifuges—one for fuel and the other for lube oil.

In the after engine room there are four more 1600-Bhp. GM-Cleveland Diesel generating sets for providing propulsion power to the port propeller. The features such as the main control board, coolers, centrifuges, pumps, etc., to be found in the forward compartment are duplicated here. But the layout is slightly different due to the four engines being together, with an auxiliary 750-Bhp., 500-Kw., GM-Cleveland Diesel generator set on either side respectively, as well as three 100-Kw. motor-generator sets, of which the motors are d.c. and the generators a.c. The four electric motors geared to the port shaft naturally are on the port side of the engine room. This engine room also functions as the main control room for the engineer-officer on watch.

The installation of auxiliary diesel generators differs, however, on some of the seven mother ships. For instance, the *Proteus*, *Nereus* and *Sperry* each have four auxiliary units of 500-Kw., and one of 60-Kw., whereas the *Bushnell*, *Howard W. Gilmore* and *Orion* have three 500-Kw. units, one 250-Kw. unit and one 60-Kw. unit. In a separate compartment on a higher level of the

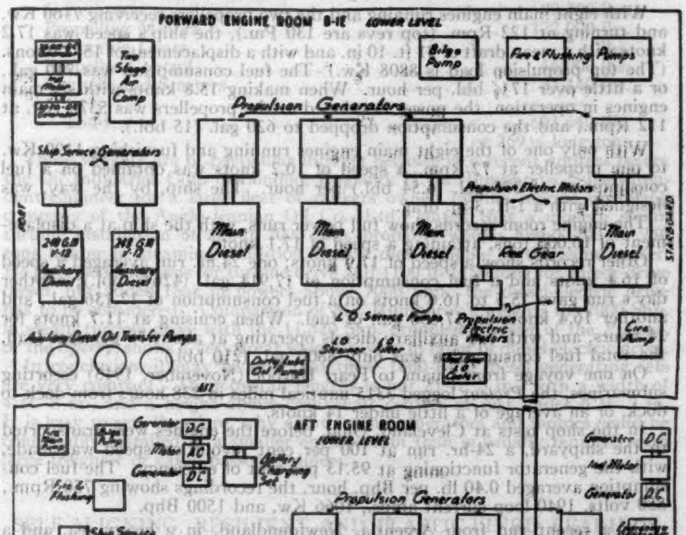
U. S. Submarine Tender "Sperry" Has Diesel-Electric Drive with Alternating Current.



View of engine room of Howard W. Gilmore, d.c. drive submarine tender. Three of the four electric motors and central reduction gear to one of the two propeller shafts can be seen. This arrangement differs from the a.c. drive of the Sperry.

Proteus there is a 90-Bhp. diesel engine turning 1200 Rpm. and driving an emergency generator to provide emergency power for vital auxiliary circuits, even if both main engine rooms are flooded or badly damaged. In the boiler room between the two main engine rooms are the two oil-fired heating boilers and six evaporators, the latter being capable of distilling 50,000 gal. of sea water per day for drinking purposes and for the storage batteries of submarines. The latter water is always redistilled to remove all traces of impurities.

A much larger engine room staff than would be needed by a merchant vessel of her size is carried on the *Proteus.* Apart from this being Navy practice because, under action, loss of life can be serious, the engineers also have the mechanical features of the entire ship to look after, including the wiring, the big cranes, heating and ventilating systems, charging of submarine batteries, etc. Altogether, 12 men per watch are in the after engine room, and 11 are



Schematic arrangement of the two engine rooms showing disposition of 12 General Motors diesel generating units.

in the forward compartment. With a passenger-cargo liner of similar power, this probably could be cut down to six men per watch, especially if all engines were installed in one compartment.

The fuel capacity of the *Proteus* is 1,200,000 gal. (3760 tons) of diesel oil, plus 77,000 gal. (248 tons) of luboil. There also is tank capacity for 255 tons of drinking water, and about 240 tons of boiler and engine cooling water. The distilled water supply for submarine storage batteries is 12,000 gal. While it will be noted that the fuel capacity is sufficient for about 65 days "steaming" at full power, in actual practice, it means that the ship has enough diesel oil to supply submarines when needed. When no "oilers" are around, she has to "feed" her dependents, as is proper with a mother.

According to data prepared by Lt. Comdr. P. E. Pontius, a former engineer officer of the *Proteus*, the following results were figured when the ship had not been dry-docked for bottom cleaning for nine months:

With eight main engines running and the two propellers receiving 7160 Kw. and turning at 122 Rpm. (top revs are 130 Pm.), the ship's speed was 17.2 knots with a mean draft of 21 ft. 10 in. and with a displacement of 15,000 tons. (The top propulsion load is 8808 Kw.) The fuel consumption was 740 gal., or a little over 17½ bbl. per hour. When making 15.8 knots with six main engines in operation, the power developed at the propellers was 5370 Kw. at 112 Rpm., and the consumption dropped to 620 gal. (15 bbl.).

With only one of the eight main engines running and furnishing 1190 Kw. to one propeller at 72 Rpm., a speed of 10.2 knots was obtained on a fuel consumption of 275 gal. (6.54 bbl.) per hour. The ship, by the way, was designed with a 1-ft. 3-in. drag.

The engine room records show full power runs, with the ship at a displacement of 16,000 tons, attaining a speed of 17.1 knots.

Other records show a speed of 17.9 knots; one 24.hr. run averaged a speed of 16.4 knots and a fuel consumption of 17,914 gal. (426½ bbl.); another day's run gave 15.5 to 16.6 knots on a fuel consumption of 17,136 gal., and another 16.4 knots on 17,650 gal. of fuel. When cruising at 11.7 knots for 24 hours, and with two auxiliary diesels operating at about 70 per cent load, the total fuel consumption was only 8800 gal. (210 bbl.).

On one voyage from Guam to Pearl Harbor (November, 1945) escorting submarines, the *Proteus* logged 3315 nautical miles in 238 hours from dock to dock, or an average of a little under 14 knots.

In the shop tests at Cleveland, Ohio, before the engines were transported to the shipyard, a 24-hr. run at 100 per cent revolution speed was made, with the generator functioning at 95.13 per cent of efficiency. The fuel consumption averaged 0.40 lb. per Bhp. hour, the recordings showing 750 Rpm., 550 volts, 1940 loop current amps., 1066 Kw. and 1500 Bhp.

On a recent run from Argentia, Newfoundland, in a heavy sea, and a 35-knot beam wind, the *Proteus* averaged 12½ knots. Both her commander and chief engineer say that she is a splendid vessel in a heavy sea, and there is no propeller racing.

It will be noticed in the comparison below that, with the a.c. drive, there is a very important saving in weight over d.c. machinery. Speed reduction is provided between the engine and motors, eliminating any need for reduction gears. This factor enables the engines to be operated constantly at higher speed and permits of lightweight design, while the propellers can be turned at lower speeds consistent with higher propeller efficiency. The first cost of a.c. drive is lower than with d.c. installation, which should be borne in mind when shipowners are asking for estimates.

On the other hand, the a.c. arrangement does not have the operating flexibility of the d.c. design, for with the latter, the propeller speed can be adjusted independently of the speeds of the diesel units. Whereas, with a.c., the main engines must be slowed down if the propeller speed is to be reduced, Also, when the propeller load is increased, the speed of the main diesel must be raised.

One important factor, in comparison with a steamship of the same power, is the weight of the machinery. This is as follows:

Submarine Tender *Proteus* with d.c. installations:	
Diesel Engines (Main)	321,400 lbs.
Reduction Gears	111,000 lbs.
Propeller Control	12,200 lbs.
Generators	134,560 lbs.
Propelling Motors	134,560 lbs.
Exciters and Starters	15,600 lbs.
Total	729,320 lbs.
	or 326 long tons
Submarine Tender *Sperry* with a.c. installation:	
Diesel Engines (Main)	321,400 lbs.
Generators	82,200 lbs.
Propelling Motors	122,900 lbs.
Reduction Gears	— lbs.
Propeller Control	46,000 lbs.
Exciters and Starters	31,140 lbs.
Total	603,640 lbs.
	or 269½ long tons

These weights are of particular interest because in a recent address before the Oil Industries Club in London, England, H. S. Humphreys, Superintendent-Engineer for a large fleet of tankers owned by the marine operating company of the Anglo-Iranian Oil Co., Ltd., stated that a modern steam turbine installation of about 7000 Shp. probably weighs about 200 to 250 tons lighter in steaming condition than a diesel engine of the same power. He gave 900 long tons for turbine drive and 1100 to 1150 tons for direct-drive diesel, but added that in powers above 7000 Shp., the diesel-electric drive has to be considered. Comparison of Mr. Humphrey's weights for 7000 Shp. in steam turbine drive with the 11,520 Shp. propulsion power of the *Proteus*, or the *Sperry*, shows a very great weight saving in favor of diesel-electric drive.

In conclusion, it may be said that these 1600-Bhp. diesel generating sets are very compact. Their overall lengths is but 21 ft. 3 in., with a width of 4 ft. 5 in., and a height above crankshaft centers of just under 6 ft. 2 in. It is easy to see, therefore, why eight engine generators and motors could be confortably installed in a smaller engine room than is usually required with large freighter or cargo-passenger liners.

SELF-ALIGNING, RESILIENT AND PLASTIC-LINED BEARINGS FOR PROPELLER SHAFTING.

This article by Mr. Robert Nelson is reprinted from the "Shipbuilder and Marine Engine-Builder" for August, 1947. It describes one application of roller bearings to propulsion shafting steady bearings—an adaptation of roller bearings which has found favor among some marine engineers.

Up to the present, the application of self-aligning roller bearings to marine machinery generally has been somewhat restricted in scope, though, in a number of ships built on the Continent prior to the outbreak of the late war, such roller bearings were fitted to support the tunnel shafting, principally with the object of reducing friction, incidental advantages afforded by bearings of this type being high radial-load capacity, together with a high value of the ratio of thrust load to radial load.

The illustration reproduced in Figure 1 shows the type of self-aligning, twin-roller bearing fitted to support the tunnel shafting in the ex-German twin-screw liner *Potsdam* (now the *Empire Fowey*).

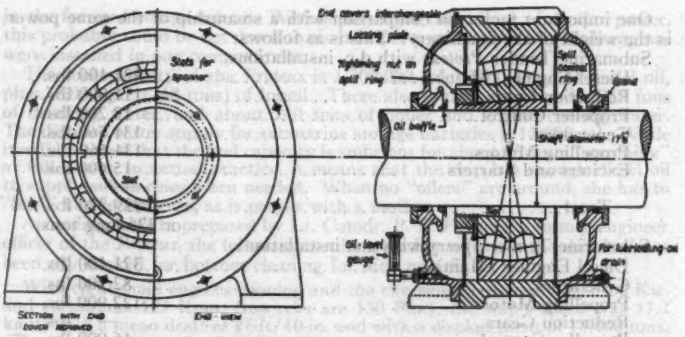


Fig. 1.—Self-aligning Roller Bearing for Tunnel Shafts.

It will be observed that the revolving, or inner, race of the bearing, which is of conical form, fits a corresponding conical split-ring, screwed at one end, and secured to the shaft and inner race by means of the round nut shown in the illustration. In general, such bearings are integral, and not in halves, a feature which necessitates the replacement of the usual flange couplings by split muff couplings at the ends of each length of shafting—an arrangement which is not new to marine engineers, since it is frequently adopted for tail-shafts to facilitate their withdrawal through the sterntube in dry-dock. The twin sets of rollers in the *Empire Fowey* appear to follow normal practice, being of the self-aligning type, and secured by lips against the inner race, while the outer, or stationary, race is a good fit in the bearing housing. To judge from the service performance of the bearings, the loss of energy due to bearing friction appears to have been exceptionally low.

The bearings run in an oil bath, having an oil-level gauge for visual inspection. The solid covers which enclose the bearing at each end are separate from the housing, and may be pushed clear, along the shaft, to afford access to the internal parts.

Mention may be made of the Timken tapered roller bearings, manufactured by British Timken, Ltd., Birmingham—a type which, by relative endwise adjustment of two opposing tapered elements, can be preloaded, as desired, to give freedom from play and to prevent vibration. In highly-developed, high-speed marine transmissions, this type of bearing is particularly suitable as a thrust bearing.

Theoretically, a thrust bearing with tapered rollers and paths is ideal, since this form gives rotation with pure rolling contact between the components. All the surfaces must, of course, be geometrically accurate; and since, even with the most accurate machining, some tolerance is required to allow for small deviations from absolute dimensions, great care must be taken. Due to the tapered construction, a purely radial load gives rise to a thrust reaction, which is generally taken up by another bearing mounted in the opposite way, and tapered roller combined radial and thrust bearings are generally mounted in pairs.

More recently, resilient bearings and plastic-lined bearings have been successfully employed—even for shafts of fairly large diameters—as alternatives to the well-known type having white-metal lining.

In regard to sterntube bearings, resilient rubber has proved a most efficient alternative to the more usual strips of lignum vitae. Figure 2 represents a part section of the after end of a typical rubber-lined sterntube. The rubber segments must be accurately fitted, so that no gaps exist between the joints. Considering the weight of the shaft, together with that of the overhanging

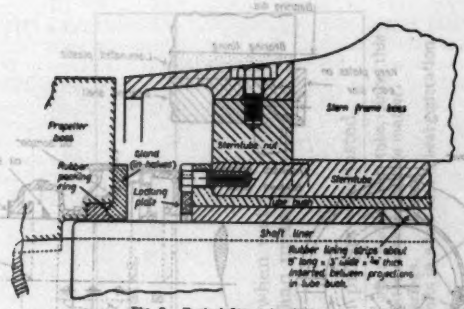


Fig. 2.—Typical Sterntube Fitting.

propeller, the compression of the rubber is indeed small—being in the region of 0.02 to 0.025 in.

Rubber-lined bearings have found favor for some time in the sterntubes of numerous types of smaller craft, *e.g.*, motorboats, coasting vessels, ferries and similar vessels, in the form of Cutless rubber bearings, which incorporate a special, soft, rubber compound, fluted or grooved to facilitate water lubrication. An important merit of rubber-lined bearings is the very low coefficient of friction of wet rubber; and, at fairly high speeds, the frictional losses associated with such bearings may be no more than those experienced with ball or roller bearings. Another advantage is that the soft, resilient lining is less liable to the injurious effects of sand or other gritty matter than is a laminated-plastic lining, since the latter is of very hard texture. Moreover, rubber absorbs shock and vibration from the shaft, and, consequently, higher operational speeds may be safely entertained. An important point is that oil and grease must not be allowed to come into contact with the rubber surfaces. Rubber-lined bearings, it may be added, have been employed for shafts of the largest diameters.

Laminated-plastic materials for the lining of tunnel and sterntube bearings have recently been adopted by one or two well-known shipping companies. The lining is in the form of strips or segments (or, in the case of short bearings, pieces of semicircular shape) of varying thicknesses, the surface being glass-hard. An essential feature of plastic-lined bearings is that adequate clearance must be allowed—of the order of 0.0035 to 0.004 inch per inch of bearing diameter in the case of sterntube bearings which have upper and lower linings.

Figure 3 illustrates one type of tunnel bearing, fitted with laminated-plastic lining, the lining being fitted to the lower portion of the bearing only, since the top portion of the bearing merely forms a cover, which serves to house the oil distributor. In passing, it may be remarked that such bearings are designed to be self-lubricating, oil-throwers and scrapers being incorporated to direct the oil into the bearings from the sump. The check plates shown in the illustration are generally secured to the casting to provide restraint for the lining.

The bearing pressure in plastic-lined bearings may be increased to something like double that generally adopted with white-metal lining; consequently, for equal diameters and loads, shorter bearing surfaces may be employed.

An important aspect of the use of plastic linings is that the lubricant may be either water (as in sterntube bearings) or oil (as in tunnel or intermediate-shaft bearings); it has been found that, with laminated-plastic sterntube bearings, the addition of grease to the water circulating in the bearing increases the life of the plastic lining. Such linings are unaffected by temperatures up to about 150 deg. C. (300 deg. F.) and are, moreover, resistant to the action of steam, oil and dilute acids.

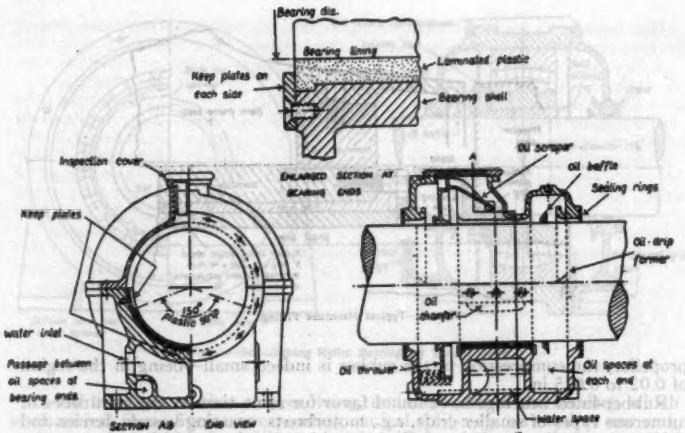


Fig. 3.—Tunnel-shaft Bearing of the Self-lubricating Type, with Laminated-plastic Lining.

The compression strength of the plastic employed for bearings is about 45,000 lb. per sq. in. at right-angles to the laminations, and about 24,000 lb. per sq. in. in the direction of the laminations; care must therefore be taken to arrange the material so that the load is applied in the appropriate direction.

General Notes on the Application of the Various Types.

A comparison of the wear, in service, of the plastic-lined sterntubes of two similar ships, engaged on different trade routes, was recently made in the United States of America. After two years' service, one bearing disclosed a maximum wear of 0.094 in., while the other showed a maximum of 0.042 in.; both sterntubes were 24 in. in diameter.

A comparison of the self-aligning twin-roller bearing, such as that exemplified in Figure 1, with its plastic-lined or white-metal lined counterpart, indicates that the roller type has a number of decided advantages. Firstly, there is no need for water cooling; secondly, no oil-throwers or scraper-rings are required; and, thirdly, with properly adjusted rollers the frictional losses are considerably smaller. The third point is especially important in roller bearings where "creep" and "skew" are liable to occur. On the other hand, these merits are somewhat offset by the higher initial cost of roller bearings, and the expert fitting they require; great care must be taken to prevent maladjustment, and this is especially important where service conditions are arduous.

No article on this subject would be complete without reference to tunnel bearings of the Mitchell type, in which (while white-metal is retained as the lining) the lined journal bush is pivoted on the bearing shell, so that the bearing may adjust itself to the variations in load which occur under service conditions. These Mitchell bearings are generally water-cooled, and incorporate oil scrapers, similar to those shown in Figure 3; thus, they are of the self-lubricating type, and, though the frictional loss associated with them is somewhat greater than that experienced with roller-type bearings, it is, in any case, small. Being in halves, the Mitchell bearings are easily accessible for inspection.

PROPELLER-SHAFT BEARINGS AND BEARING MATERIALS.

Type of Lining or Bearing.	Application.	Lubricant.	Remarks.
White-metal.*	Tunnel and sterntube.	Oil.	Existing type.
Lignum vitae.†	Sterntube.	Water.	Existing type.
Rubber.	Sterntube.	Water.	Low frictional losses only when wet.
Laminated plastic (*e.g.*, Formapex Miocarta).‡	Tunnel and sterntube	Oil, grease or water.	Much lower rate of wear than metallic bearings.
Roller bearing.	Tunnel.	Grease.	Low frictional losses when properly adjusted.

*In white-metal lined sterntubes, an oil gland, of the Newark or Vista type, is generally incorporated at the after end of the tube, since this alloy must be lubricated by oil. These devices retain the oil in the bearing, and at the same time exclude sand, water or other foreign matter.

†Lignum vitae, being a wood of great hardness and strength, is particularly suitable for use in under-water bearings.

‡Formapex Miocarta laminated plastic has a low thermal conductivity, but the small coefficient of friction results in a low rate of heat generation.

One important British shipping company, who have had a certain amount of service experience with plastic-lined tunnel bearings, state that, up to the present, no adverse report has been received on their behavior or performance. The material used was Formapex Miocarta, moulded in semicircular form, and machined to suit the bearing shell. The makers of this material state that the arc of contact should not exceed 150 deg.; and, within certain limits the greater the original thickness of the lining, the more economical, in general, it proves to be. Formapex Miocarta is manufactured from specially woven textile fabrics, compounded with synthetic resin and compressed at high temperature and high pressure into suitable form, and machined, where necessary, to the bearing dimensions.

In conclusion, it may be mentioned that, while water-lubricated rubber linings have been successfully employed in sterntubes, they have not been adopted for tunnel-shaft bearings. Great care must be taken during installation of rubber-lined bearings to secure correct alignment, which is an essential feature; moreover, provision must be made for ample and continuous water lubrication in service.

The table has been compiled to summarize the various types of bearings and bearing materials. The first two represent standard normal practice, but it is clear that the interesting alternatives have before them a wide field of application, and are probably best judged from actual service experience.

STRUCTURAL FAILURE OF A RIVETED SHIP.

The following article by Mr. E. M. MacCutcheon, Naval Architect, U. S. Coast Guard, is reprinted from "The Welding Journal" for January, 1948.

The *Oakey L. Alexander* was carrying a load of coal from Norfolk, Va., to Portland, Me. A severe storm lashed the Atlantic coast. The heavy seas caused distress to shipping and was too much for the 31-year-old ship. Two huge waves boarded the ship from the starboard quarter, washed overboard the port lifeboat, damaged the starboard lifeboat and the boat deck, besides bashing in the oak doors in the deck house. The load of the seas was too heavy. The bow broke off. Her captain grounded her on the rocky coast near Portland, Me.

When the seas had calmed the ship was boarded to investigate the structural failure and compare it with structural failures of welded ships.

During the last four years, the problem of structural failure in welded merchant ships was the subject of an intensive investigation. Studies were made of all phases of ship construction procedures and a thorough research program was carried out in an attempt to discover the basic cause of hull fractures.

Some of the more common questions that have been asked concern the difference in the relative performance of ships, constructed by means of riveting or welding. There are some facts which throw light on this difference. The following data form the basis for a partial answer to the question: "Were similar difficulties experienced on riveted ships?" Structural failures are not unknown on riveted ships. Such famous ships as the *Leviathan* and *Majestic* cracked their main decks. In other cases, such as the *Oklahoma* and *Mielero*, the ships broke in two.

These cases are well known in marine circles but the numerous lesser fractures have gone unheralded and without record. Attempts to compile statistics based on such minor failures are thwarted by the lack of evidence. On the other hand, for the last four years, the structural-failure data on welded ships were kept in a single central record where they were tabulated and analyzed.

FIGURE 1.—DAMAGED FREIGHTER "OAKEY L. ALEXANDER" GROUNDED AT CAPE ELIZABETH, ME.

FIGURE 2.—STARBOARD SIDE LOOKING FORWARD AT EDGE OF CURLED-UP DECK PLATING. THE CLEAVAGE FRACTURE TURNS TO SHEAR IN THE VICINITY OF RIVETED SEAM AND THEN BACK TO CLEAVAGE BEYOND THE SEAM.

FIGURE 3.—COAMING OF NO. 4 HATCH, STARBOARD SIDE LOOKING OUTBOARD, SHOWING THE MAIN DECK CURLED UP UNDER THE FORCE OF DAMAGE.

During the last war, great numbers of nearly identical welded ships were constructed. Upon the basis of the service records of these similar structures, it was possible to prepare statistical studies. No similar volume of production of riveted ships ever took place. Even the shipbuilding program of Hog Island ships during World War I did not approach the production volume of Liberty ships and T-2 tankers during World War II. The large volume of production of welded ships and the broad survey of structural failures made during the last four years permitted a complete statistical analysis which was presented in the final report of the Board to Investigate the Design and Methods of Construction of Welded Steel Merchant Vessels. Since similar records are not available with respect to riveted ships, the exact answer to the question of their relative performance will never be obtained.

Even before the *Oakey L. Alexander* broke in two, many people asked the question: "Were the structural failures of welded ships similar to those which occurred on riveted ships?" This question could not be answered for a long time because few facts were available for detail study or ready comparison.

The many investigations made on the structural failures of welded ships, broadened the knowledge of fracture characteristics of steel plates and of typically sensative design features.

When the *Oakey L. Alexander* was boarded by structural experts, they applied the knowledge gained in four years of investigation and analysis of welded-ship structural failures. It was possible to survey this ship and compare the findings with the results of investigations of other ship failures. All the phases of the hull structural failure were carefully photographed, sketched and measured. Various samples were removed and forwarded to the National Bureau of Standards for analysis. The final results of this survey will not be available for many months. The general observations made at the survey are very interesting and will in part answer the question quoted above.

The fractures were found to have started at the square corners of the hatches. In addition to the main fractures which broke the ship in two, two minor fractures were found at other hatch corners. These fractures, however, did not continue to spread.

The fracture of the plates was classified as "cleavage." The term cleavage is applied to a crystalline-appearing fracture which is square to the plate surface, exhibits little reduction in thickness or stretching of the material, and which usually has a discernible herringbone pattern pointing toward the starting point of the fracture.

In the way of the starboard deck seam and at certain other locations, it was found that the mode of separation changed from "cleavage" to "shear." A "shear" fracture is silky in appearance, occurs at 45° to the plate surface and is usually accompanied by considerable stretching of the plate. Such fractures were rarely found in the welded ship failures except where a fracture terminated. In this ship, it would appear that the fractures had run from the hatch corners to the riveted seams where a slight pause took place and the fracture changed to "shear." The pause was only temporary, however, as the load on the riveted seams was more than the plating could stand and ultimate failure of the entire hull resulted.

All characteristics of the welded steel ship failures were present. Certainly, it can be said that the structural failures of the two types of ship are essentially the same. The presence of the riveted seams undoubtedly arrested the spread of cracks in numerous cases of riveted ship failures. It is likely that this resistance to the continuation of cracks is the one real difference between the two types of structures. Cracking of one or two plates is not spectacular and would not be expected to attract broad interest. Certainly, the modes of fracture are not fundamentally different and it is likely that the more illusive features attributed to the riveted structure are due to nothing more than lack of detailed facts regarding structural failures in the riveted hulls.

ALUMINIUM ALLOY BEARINGS IN DIESEL ENGINES.

The following article by Mr. D. B. Wood of the Aluminium Co. of America is reprinted from "The British Motor Ship" for September, 1947.

Experimental work on the development of aluminium alloys for bearings was begun by the Aluminum Company of America in 1936. Prior to this date some development work on aluminium bearing alloys had been completed in both England and Germany, with the result that some data and experience had been accumulated on the performance of these materials in test engines as well as those in actual service.

The program was initiated with its primary goal being the development of an aluminium alloy that would have excellent bearing characteristics, as measured by known standards and as evlauated on testing machines in use for such work. At first, emphasis was placed upon obtaining the desired bearing characteristics rather than on the strength of the material itself. Altogether several hundred alloys were prepared and screened for further testing. The result of this investigation was the development of the 750 type of alloy.

The following table gives the mechanical properties of 750 alloy, which is now available in two tempers:—

	750-T5	750-T101
Tensile Strength p.s i...	22,000	23,000
Yield Strength (Tension) p.s.i. ..	10,000	16,000
Yield Strength (Compression) p.s.i...	10,000	16,000
Shear Strength p.s.i.	14,000	—
Endurance Limit p.s.i...	9,000	—
Elongation, per cent.	12	8
Brinell Hardness	45	50
Rockwell Hardness ("H")	75	85
Density	2.88	2.88
Thermal Conductivity..	0.44	0.44
Coefficient of Thermal Expansion per °F..	.0000135	.0000135

Nominal Composition: Tin 6.5%, Copper 1%, Nickel 1%, remainder commercially pure aluminium.

The "T5" (formerly T533) temper of alloy 750 indicates that the material has been given a stabilizing heat treatment after casting, whereas the "T101" temper signifies that the casting has been given 4 per cent cold work after a stabilizing heat treatment. As laboratory data and test experience in the field accumulated, it soon became evident that alloy 750 in the stabilized condition would need additional strength where high operating temperatures existed. This led to subsequent cold workings, which increased the yield strength about 6000 p.s.i.

As the different alloys were tested for resistance to scuffing, it was found that those with silicon additions showed considerable promise. Further experiments along this line resulted in the release of alloy XA750 on an experimental basis.

The mechanical properties of XA750, also available in two tempers, are given in the following table:—

	XA750-T6	XA750-T91
Tensile Strength p.s.i.	22,000	23,000
Yield Strength (Tension) p.s.i. ..	10,000	17,500
Yield Strength (Compression) p.s.i.	10,000	17,500
Shear Strength p.s.i...	14,500	—
Endurance Limit p.s.i.	9,500	—
Elongation, per cent...	10	7
Brinell Hardness	45	50
Rockwell Hardness ("H")	75	85
Density	2.83	2.83
Thermal Conductivity	0.40	0.40
Coefficient of Thermal Expansion per °F.	.0000132	.0000132

Nominal Composition: Similar to alloy 750 except for addition of silicon.

In this alloy, the symbol "T6" (formerly T7) is used to represent material that has been given a solution heat treatment, while the symbol "T91" indicates that the alloy has been heat treated and then cold worked 4 per cent.

The cold working of alloys 750 and XA750 may be accomplished by several methods. The two most practical are: (1) To squeeze the unconfined casting between the jaws of a press, thereby reducing its length by 4 per cent. (2) To pass the casting through a die and reduce the outside diameter a given amount. The first method is the more economical when different sizes of castings are to be cold worked. Test specimens taken from different points in castings subjected to this type of cold working show that it produces a surprisingly uniform structure.

Although test work in the laboratory indicated that alloy XA750 was more resistant to scuffing and scoring than alloy 750, it was not deemed advisable to recommend the use of alloy XA750 with soft journals because of its silicon content. Where journal hardness is over 300 Brinell, however, alloy XA750 may be used to advantage, particularly if doubtful lubrication conditions exist.

The war curtailed development work and tests, and it was not until 1944, when restrictions on the use of aluminium were lifted, that this activity was resumed. Commercial production did not begin until about January, 1945.

Practically all Diesel-engine builders have tested or are now testing samples of aluminium alloy bearings. Although most of this test work has been successful, the few failures that have occurred in laboratory and field tests can be attributed in every instance either to improper installation or lack of understanding of the design requirements for aluminium alloy bearings.

Most of Alcoa's commercial production of bearing alloys has thus far been in alloy 750, with nearly all bearing material being furnished in the form of permanent mould castings. Manufacturers of aluminium alloy bearings include: American Bearing Co., Indianapolis, Indiana; Cleveland Graphite Bronze Co., Cleveland, Ohio; Federal-Mogul Corporation, Detroit, Michigan; and Johnson Bronze Co., New Castle, Pennsylvania.

Those characteristics that make aluminium alloy bearings so adaptable for Diesel-engine use are:

(1) Ability to carry heavy loads. In several tests where the effective area of the bearing has been reduced in order to increase the unit bearing load to very high values, aluminium alloy bearings have performed satisfactorily with loads ranging from 7500 to 8000 p.s.i. In fact, some bearings were found suitable with loads approaching 10,000 p.s.i. In this range, however, every detail must be correct to obtain satisfactory performance. Normally, a load of 5000 p.s.i. may be taken as the maximum figure at which successful opera-

tion should be expected, unless there are details in the mechanism which relieve or help those conditions that usually limit load-carrying ability.

(2) Aluminium alloy bearings have good plasticity. In use, particles of dirt may become embedded in the surface of aluminium alloy bearings. The degree to which these particles will embed and the quantity taken up will probably depend upon the relationship between the hardness of the shaft and that of the bearing. Aluminium alloy bearings will conform to any normal misalignment of engine parts, and they can be scraped in by conventional methods to conform to irregularities of contacting surfaces. In this respect aluminium alloy bearings are probably better than any other bearing material except babbitt.

(3) Aluminium alloy bearings are highly resistant to adherence with adjacent parts. In the event of failure, aluminium alloy bearings will not damage the journal, except possibly in cases where seizure has been very rapid and high localized stresses have been encountered. Because of the relative softness of aluminium alloy bearings, scratches are easily made in their surface. However, they do not tend to pick up or scuff unless some other unfavorable conditions exist.

(4) Because of the metal's ability to transmit and dissipate heat rapidly, aluminium alloy bearings will normally operate at lower temperatures than bearings made of other metals. For this same reason there is a tendency for local points of high temperature to average out much more quickly.

(5) Aluminium alloy Diesel-engine bearings are fabricated from a single piece of material. Therefore there are no heat barriers at the bond line, or bonding layers which may fail in service through fatigue.

(6) Aluminium alloy bearings are highly resistant to corrosion from oils in use today. In addition, they do not react with these oils to form harmful products.

(7) Aluminium alloy bearings, because of their basic simplicity, should be low in cost when compared with the more intricate duplex or multiplex products designed to do a similar job.

To obtain the full benefit of the advantages offered by alunimium alloy bearings, however, consideration must be given to the following factors in designing and installing the bearings:—

(A) Aluminium has about twice the coefficient of thermal expansion of ferrous materials. It is essential, therefore, that ample oil clearance be provided along with closely controlled crush and sufficient end clearance.

Oil clearance should always be at least .001 in. per inch of journal diameter (preferably .00125 in.). This is a minimum value, and all dimensional tolerances should apply to the plus side. In case of very small diameter bearings, a minimum of .002-in. total oil clearance should not be violated. This ample oil clearance probably serves a twofold purpose, in .that it allows for the inward expansion of the bearing as it is brought from room temperature to operating temperature, and also provides a free passage for dirt particles.

In considering crush, tests have shown that a load of 2000 p.s.i. on the parting-line area of the bearing will normally give good contact between the back of the bearing and its housing. Using this figure of 2000 p.s.i. as an inspection load, manufacturing tolerances of the bearing itself add to actual parting-line joint crush, as does also the use of a bearing checked in a given fixture and used in a smaller-radius engine bore. It is obvious, therefore, that manufacturing tolerances of the bearing and the housing should be held to the closest practicable limits in order to control crush adequately. It seems illogical that the same tolerance should be specified or maintained regardless of bearing diameter or length. A table such as the following may be helpful in controlling this situation:—

This would provide larger manufacturing limits as the diameter of the bearing increases, and would tend to keep the actual stress in the bearing shell (caused by crush) more evenly distributed throughout a range of bearing sizes. Stress in the bearing shell resulting from manufacturing tolerances on

Tolerance on circumferential length of bearing in fixture of radius = 1/2 maximum bore. Loaded to 2,000 p.s.i. parting line area.

Tolerance (Inch)	Journal Diameter (Inches)
.0000-.0010	1-1/2- 3
.0000-.0015	3- 5
.0000-.0020	5- 7
.0000-.0025	7- 9
.0000-.0030	9-12
.0000-.0035	12-20

the bore should be calculated, and added to the stress caused by manufacturing tolerances of the bearing itself and the inspection load stress.

Another factor that affects crush stress is the taper of the parting-line face of the bearing. Usually this taper is expressed as a certain tolearnce regardless of the length of the bearing. The following table may possibly give more uniform results:—

Tolerance allowable on taper of parting line face from plane of face of fixture of radius 1/2 maximum bore diameter.

Tolerance (Inch)	Bearing Length (Inches)
$\pm$ .00025	0-3
$\pm$ .00050	3-6
$\pm$ .00075	6-9
$\pm$ .00100	9 and over

Naturally, the manufacturing tolerance on bearing wall thickness increases with bearing diameter. Spread at the parting line may be critical with aluminium alloy bearings if limits normally used with steel-backed bearings are adhered to. With aluminium alloy bearings spread at the parting line should be held to .000 in. to .005 in. This figure gives excessive stress in some cases, but it may be the limit to which bearings can be held in quantity production. Anything less than this upper limit of .005 in. will help to keep bending stresses at a proper level.

(B) Engines should be reasonably clean before starting. In order to obtain trouble-free operation, extreme care should be exercised to see that the entire engine is clean internally. It is difficult to understand why more time is not spent in cleaning a mechanism which represents so large an investment in material, labor and engineering. Oil filters and strainers of adequate capacity should be provided, but they should not be used in lieu of thorough cleaning. Many bearing failures can be attributed to dirt left in the engine, and the resulting damage may make the engine a problem throughout its life.

(C) Some provision should be made to provide positive lubrication at the instant the engine starts. While aluminium alloy bearings will live under very

adverse conditions, no true evaluation has yet been made of the critical initial start. Oil pumps of ample capacity should be provided, and a large quantity of oil should be circulated throughout the engine.

(D) Diesel engines do not normally operate with sump oil temperatures that are high enough to be critical with respect to aluminium alloy bearings. There are instances, however, where existing high temperatures could be corrected or controlled through the use of adequate oil coolers capable of handling a large quantity of oil. This may be particularly important where an engine would be called upon to maintain maximum horsepower for an extended period of time.

(E) Care should be exercised in handling aluminium alloy bearings so that nicks and gouges will not result.

Where conditions are not favorable to trouble-free operation, a corrugated bearing surface (in the direction of rotation) and tin or babbit plating of the running surface may be beneficial. Aluminium alloy bearings should not be expected to support abnormal loads, such as may result if shims are used in machining parts and then removed in installation.

Where a bearing is required to meet higher service temperatures in Diesel engines, another aluminium alloy has been developed. This alloy, called XB750, has the following approximate physical properties:—

	XB750-T5 (Formerly T533)
Tensile Strength p.s.i.	30,000
Yield Strength (Tension) p.s.i. ..	20,000
Yield Strength (Compression) p.s.i. ..	20,000
Elongation, per cent.	5
Shear Strength p.s.i.	21,000
Brinell Hardness	70
Rockwell Hardness ("H")	100
Density	2.88
Thermal Conductivity	0.43
Coefficient of Thermal Expansion per °F.	.0000133

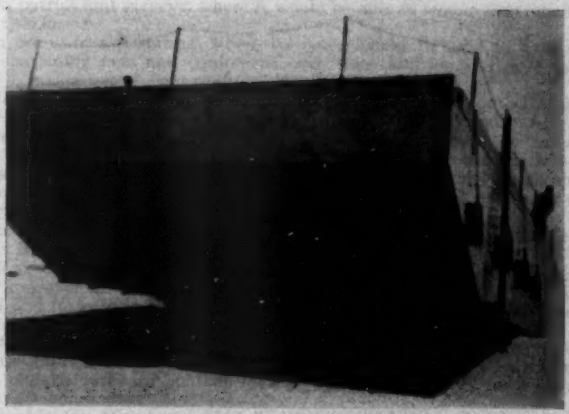
Laboratory tests indicate that XB750 will retain interference fit or crush up to about 300 degrees F. when proper initial crush allowance is made. This bearing alloy should not be used in conjunction with journals that have a Brinell hardness of less than 300, nor should it be used where much dirt is present or where alignment and tolerances of engine parts are not good. Tests results indicate that XB750 alloy parts show a performance comparable with that of bronze in many bushing applications.

Although the entire Diesel-engine industry is rapidly gaining experience with aluminium alloy bearings, it will be some time yet before all the experience and data already collected can be evaluated. It can be fairly stated, however, after two years of actual use and nine years of laboratory and engine tests, that aluminium alloy bearings give promise of excellent service at a reasonable cost in Diesel engines.

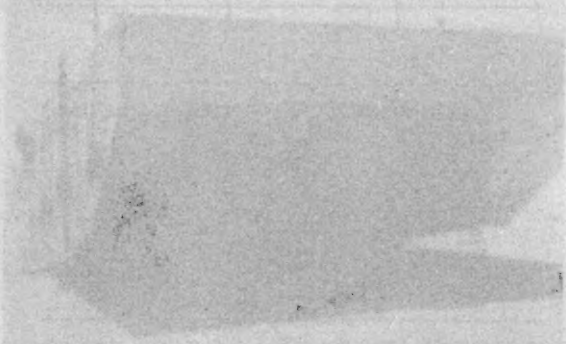
THE "WALKING BARGE."

The following article by Mr. Warren E. Young, describing a novel type of amphibious craft, is reprinted from "The Military Engineer" for January, 1948.

A new amphibious vehicle, the "Walking Barge," which can carry 60 tons of men and material over sand, soft and hard ground, oozy-to-liquid mud,



THE "WALKING BARGE."



tidelands, surf and ocean, and can climb steep embankments and traverse uneven terrain, all in a sort of leapfrog motion, has been developed by the Navy's Bureau of Yards and Docks at the Advanced Base Depot Proving Ground, Port Hueneme, California.

In addition to its present functions, recommendations have been made for testing the barge for the following uses:

1. A carrier for sectional road mats which may be laid behind it as it heaves itself out of the surf and walks inland.
2. A carrier for vehicles which may run down the road matting as it is laid.
3. A stable, and at the same time mobile, platform for use in construction of causeways, bridges, docks and piers, and in placing anchorages and moorings.
4. In salvage operations where beached landing craft and ships have to be moved out of surf or off a beach where other amphibious vehicles can not operate successfully.
5. A mobile fortress which, when given additional armor, might be of value in covering foot troops during landing operations.
6. On patrols and surveys, where it would be able to reconnoiter regardless of the terrain.
7. In arctic transportation, where its power could be used to haul snow trains containing men and equipment over snow and ice and over tundra turned to quagmire.

Of all-welded construction, the barge consists of three pontoons placed side-by-side. The two outboard pontoons, each 60 feet long, are 6 feet wide and 9 feet 2 inches high. Between them is the inboard pontoon, 16 feet wide, 44 feet long, and 6 feet ¼ inch high. These hulls are framed with 4-inch structural steel angles. Plating covering the frames is ⅜ inch thick on the sides and ¼ inch on the bottom. The pontoons are divided into water-tight compartments.

On land, the barge moves in a sort of leapfrog motion. First the inboard pontoon is lifted 17 inches off the ground, moves forward 10 feet at this level and is lowered to the ground. It is followed by the outboard pontoons, which are decked over and carry the payload. This part of the structure surges upward and forward in the same manner as the inboard pontoon and comes to rest in its original position over the inboard pontoon.

Te create a lever action which assists in reducing the initial force required to lift the outboard pontoons in the first stage of a forward step, the stern bottom of each outboard pontoon is flared upward 7½ inches for a distance of 9 feet 10 inches.

Powered by two Diesel-type M4A-2 tank motors, the barge has first, second, and reverse gears. The internal mechanism which performs the walking motion may not be described for security reasons.

In appearance, the "Walking Barge" on land may be likened to a turtle, but its action suggests the movements of a frog humping itself cautiously forward.

Traction in mud flats in the present model is obtained through vertical fins installed in the bow of the inner pontoon and the forward sections of the outboard pontoons. These fins are raised automatically into the hull as the barge makes a forward "step" and are lowered into the mud when it touches.

For travel in water, a power take-off from the engine drives a propeller located in the stern of the inboard pontoon. This propeller-and-drive arrangement is designed so that the propeller can operate at various angles, and when not in use is housed in a well, out of the way for land travel. When entering water, the barge walks out as far as possible before the propeller takes over. To make a landing, the barge is driven as far as possible onto the beach with the propeller and then is walked ashore.

For steering on land, steering pads controlled by hydraulic cylinders, have been built into the stern of each outboard pontoon. Used singly, they act as a pivotal drag which causes the barge to turn.

The first working model, designed by Lt. Comdr. J. T. Tucker, United States Naval Reserve, was built at Pearl Harbor near the end of the war. It was constructed of ordinary Navy lighter pontoons and copied from the scale model with which preliminary tests were conducted in mud puddles made for the purpose. The first working model was powered by a D8 tractor set in the inboard pontoon.

Now a civilian, Commander Tucker is superintendent at the Advanced Base Depot Proving Ground, Port Hueneme, where further tests are being conducted with the barge.

The Bureau of Yards and Docks is being represented at these tests by Comdr. J. J. McGaraghan, Proving Ground Officer and Lt. Comdr. A. P. Pratt, United States Naval Reserve (Inactive), assistant superintendent.

Capt. B. W. Fink, Officer-in-Charge of the Advanced Base Depot, has aided materially in conducting the tests.

HEARING STRESS DISTRIBUTION IN BOX GIRDERS WITH MULTIPLE WEBS.

The following article by Mr. A. S. Neiman of Vacuum Concrete, Inc., reprinted from the Proceedings of the American Society of Civil Engineers for February, 1948, contains an interesting discussion of the determination of shearing stresses in a ship girder.

Synopsis.

A method for finding the distribution of shearing stress in box girders with multiple webs is presented in this paper. In effect, the structure is allowed to distort under an assumed set of shearing forces. A correction is then introduced to satisfy conditions of geometry known to exist. The particular example solved indicates that in some parts of the structure the shearing stresses are considerably greater than the average.

Introduction.

In computing the shear stress in a box girder with multiple webs, the formula frequently used is

$$v = \frac{VQ}{It} \qquad (1)$$

in which v is the unit shear stress at any point; V is the external shearing force; Q is the statical moment of the sectional area above the plane considered, about the neutral axis; I is the moment of inertia of the area of the cross section; and t is the combined thickness of members cut at the section. Eq. 1 gives the shear stress for a beam with single web of thickness t, or for a symmetrical box girder section with two webs of combined thickness t. For a girder with more than two web plates, such as is used frequently in heavy grillage foundations, or for a ship with longitudinal bulkheads, Eq. 1 is inadequate because it gives an average shear-stress value for all web members at the section. The actual shear stress may vary considerably from the average value. This is particularly true where web members are not of equal thickness, as is frequently the case for ships with longitudinal bulkheads.

The distribution of shear stress throughout a multiple-web beam, such as a ship with longitudinal bulkheads or a vessel with bulkheads that do not extend the full height, can be obtained by the mechanics of "shear flow" and the application of the principles of continuity to contiguous cells. The term "shear flow" as used herein denotes shear force per unit of linear measurement. Therefore, the shear flow at a point is the shear stress multiplied by the thickness of the metal at the point. The concept of shear flow is used extensively in the design of airplane structures.* The cross section of a vessel under

*"Airplane Design," by K. D. Wood, Purdue Univ., Lafayette, Ind., 1941.

external shearing forces is subjected to shearing stresses which may be considered as "flowing" through the section. This shear flow must be in equilibrium with the external shear force at the section, or with a combination of the external shear and the shear resulting from twisting moments. Moreover, the shear distribution must be such that contiguous cells undergo the same angular distortion. Therefore, the solution of the problem of shear distribution in a complex structure such as a ship with three or more shear-carrying webs, or with multiple decks, requires the application of the principles of geometry and statics. Two cases of shear distribution will be considered.

NOTATION.

The following letter symbols adopted for use in this paper, and in its discussions, conform essentially with American Standard Letter Symbols for Structural Analysis, prepared by a committee of American Standards Association with Society participation:

A = area of one or more cells in the cross section of a tanker; ΔA = an increment of area of a web, or member = $t\,\Delta L$;
F = "function of"; F(I) and F(II) are simplifying substitution quantities defined by the left-hand side of Eqs. 4*a* and 4*b*, respectively;
G = modulus of elasticity in shear;
I = moment of inertia of the area of the cross section;
L = length of a web, or member; ΔL = an increment of length;
M = bending moment; M_t = external twisting moment;
Q = statical moment of the sectional area above the plane considered, about the neutral axis;
q = shear flow, in pounds (or kips) per unit of length:
q_a and q_b = shear flows introduced into cells I and II;
q_{BC} = shear flow at point B in member BC;
q_{GH} = shear flow at point G in member GH;
q_0 = known shear at any point;
q_1 = assumed shear flow;
q_1' = average assumed shear flow;
t = thickness of a web or combined thickness of the members cut at a section;
V = external shearing force;
v = unit shear stress at any point;
z = moment arm; distance from the neutral axis to the centroid of the increment of area (ΔA) bounded by the plane upon which the shear is known and the plane upon which the value of shear is desired; and
θ = angle of twist.

CASE I. NO TORSIONAL MOMENT AT SECTION.

If no twisting moments exist at the section, the shearing stress must be the result of increments of longitudinal flexure only. The procedure for determining the distribution of shear stress for this case can be described in four steps:

1. Assume a distribution of shear flow q_1 consistent with the formulas:

$$q_1 = \frac{V}{I} \Sigma z\,\Delta A \quad \ldots\ldots (2a)$$

or

$$q_1 = q_0 \pm \frac{V}{I} z\,\Delta A \quad \ldots\ldots (2b)$$

in which V is the total shear at the section; I is the total moment of inertia about the neutral axis; q_0 is a known shear at any point; and $z\,\Delta A$ is the statical moment, about the neutral axis of the increment of area bounded by the plane upon which shear is known and the plane upon which the value of shear is desired.

2. Compute

$$2\,A_1\,G\,\theta = \Sigma q_1 \frac{\Delta L}{t} \quad \ldots\ldots (3)$$

in which A_1 is the area of one cell; G is the modulus of elasticity in shear; and $\frac{\Delta L}{t}$ is the ratio of increment of length to corresponding thickness. Eq. 3 gives the angle of twist, θ, under the assumed shear flow, q_1, in each cell.

3. Since in this example no torsion is acting on the structure, the angle of twist for each cell equals zero. Hence a shear flow acting around each cell can be introduced so that the resultant angle of twist of the cell is zero. This procedure can be followed for each cell by the use of equations similar to Eq. 3. For example, for a two cell structure (Fig. 1), the shear flows q_a and q_b are

FIG. 1

introduced into cells I and II, respectively, as corrections to the assumed shear flow q_1 so that the resultant angle of twist will be zero.

Hence the following formulas are obtained:

$$q_a \Sigma \frac{\Delta L}{t}\begin{bmatrix} a \rightarrow b \\ b \rightarrow c \\ c \rightarrow d \end{bmatrix} + (q_a - q_b)\frac{\Delta L}{t}[d \rightarrow a] - \Sigma q_1 \frac{\Delta L}{t} \text{ (for cell I)} = 0 \ldots . (4a)$$

and

$$q_b \Sigma \frac{\Delta L}{t}\begin{bmatrix} d \rightarrow e \\ e \rightarrow f \\ f \rightarrow a \end{bmatrix} + (q_b - q_a)\frac{\Delta L}{t}[d \rightarrow a] - \Sigma q_1 \frac{\Delta L}{t} \text{ (for cell II)} = 0 \ldots . (4b)$$

in which the symbol [a $\rightarrow$ b], etc., denotes reference to the member extending from point a, Fig. 1, to point b.

If by symmetry, or by judicious choice of the assumed shear flow, q_a or q_b is equal to zero, Eqs. 4 would be reduced to a single equation:

$$q_a \text{ or } q_b = \frac{\Sigma q_1 \frac{\Delta L}{t}}{\Sigma \frac{\Delta L}{t}} \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (5)$$

4. From Eqs. 2 and 5, denoting q_a or q_b as q_2, the resultant shear flow at any point is found equal to

$$q_3 = q_1 + q_2 \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (6)$$

CASE II. WITH TORSION AT SECTION.

The shear distribution throughout a beam section will be modified from that of case I if the beam is subjected to twisting moments in addition to flexural moments. A solution of this problem may be obtained by a procedure similar to that for case I except the steps 3 and 4 should be modified.

In a continuous structure the angles of twist of any two contiguous cells (see Fig. 1) are equal. Therefore, the left side of Eq. 4*a*, step 3, say, $F(\text{I})$ must be equal to the left side of Eq. 4*b*, say $F(\text{II})$; thus:

$$F(\text{I}) = F(\text{II}) \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (7a)$$

Another equation is obtained from statics by equating the moment of the internal shearing forces with the external twisting moment, M_t, with the result that

$$F(\text{I}) + F(\text{II}) = M_t \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (7b)$$

This procedure gives two equations from which the unknown q_a and q_b can

be determined. The final step giving the resultant shear flow at any point is obtained by combining q_a or q_b, with q_1, as indicated in step 4, case I.

The signs of the various quantities $q\frac{\Delta L}{t}$ in Eqs. 4 to 7 may be kept consistent by assigning positive or negative signs to these quantities according to whether they cause clockwise or counterclockwise rotation in a particular cell.

EXAMPLE.

Fig. 2 shows a cross section of a tanker subjected to a vertical shear of

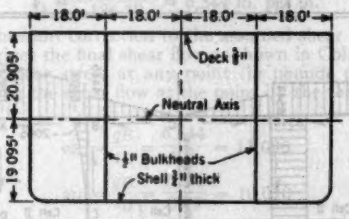


FIG. 2

12,000 kips. The moment of inertia about the neutral axis is 635,000 in².-ft². It will be assumed that the torque at the section equals zero.

Table 1 indicates, in an orderly manner, the arithmetical computations necessary to obtain the shear distribution for this example. Col. 5, Table 1, gives a set of assumed values of shear flow q_1 computed by Eq. 2*b*. These values of shear flow are plotted in Fig. 3. Because of symmetry the shear flow at points C and H equals zero; hence the shear flow (in pounds per linear inch) at point B in member BC is obtained directly from Eqs. 2:

$$q_{BC} = \frac{V}{I}\Sigma\,\Delta A = \frac{12{,}000{,}000}{635{,}000}\,20.905 \times 18 \times \tfrac{3}{8} = -4{,}445$$

and, similarly,

$$q_{GH} = \frac{V}{I}\Sigma\,\Delta A = 18.9 \times 19.095 \times 18 \times \tfrac{1}{4} = -4{,}872$$

The sign of shear flow is negative since the rotation is clockwise.

TABLE 1.—SHEAR DISTRIBUTION, TYPICAL MIDSHIP SECTION OF A TANKER

Member[a] (Fig. 3)	ΔA (ft-in.)	z (ft)	$z\,\Delta A$ (ft²-in.)	q_1 (Lb per in.)	q'_1 (Lb per in.)	$\frac{\Delta L}{t}$	$q'_1\frac{\Delta L}{t}$	q_s shear flow[b]	Shear stress[c]
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
E	....	....	....	0	....	....	....	6,544	13,090
B(E)	10.452	10.452	109.24	− 2,065	− 688	41.81	− 28,765	4,479	8,960
B(C)	11.250	20.905	235.2	− 4,445	....	....	....	−4,445	7,110
B(A)	....	....	....	− 6,509	− 8,732	28.8	−251,482	34	50
A(B)	11.250	20.905	235.2	−10,954	....	....	....	−4,410	7,060
A(D)	15.080	10.452	163.9	−10,954	−13,020	....	−363,649	−4,410	5,880
D	....	....	....	−14,052	....	....	....	−7,508	10,010
F(D)	14.32	9.547	136.7	−11,469	−13,190	25.46	−335,817	−4,925	6,570
F(G)	....	....	....	−11,469	....	....	....	−4,925	6,570
G(F)	13.50	19.095	257.8	− 6,596	− 9,032	....	−216,768	− 52	70
G(H)	13.50	19.095	257.8	− 4,872	....	....	....	−4,872	6,500
G(E)	9.547	9.547	91.2	− 1,723	− 574	38.19	− 21,921	4,820	9,640
Σ						186.19	−1,218.402		

[a] Member or point; the designation B(E), for example, signifies the shear in member BE, Fig. 3, or the properties of member BE. [b] Shear flow, in pounds per inch. [c] Shear stress, in pounds per square inch.

TABLE 2.—COMPUTATION OF SHEAR FLOW AROUND CELL II, FIG. 3, IN POUNDS PER LINEAR INCH

Shear flow (1)	Member (2)	Computation (3)	Assumed flow (4)	Total flow (5)
q_E	....	q_0 in Eq. 2b	0	6,544
q_B	BE	18.9×0.5×20.902×20.905×0.5	− 2,065	4,479
q_B	BA	−2,065+(−4,445)	− 6,510	34
q_A	AB	−6,510−(18.9×20.905×18×5/8)	−10,954	−4,410

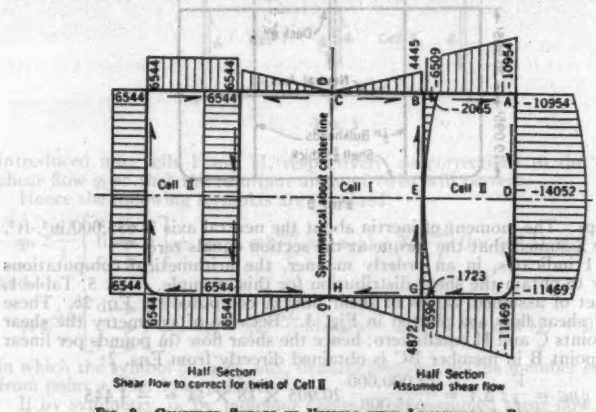


FIG. 3.—COMPUTED SHEARS IN VESSELS WITH LONGITUDINAL BULKHEADS, IN POUNDS PER LINEAR INCH (CLOCKWISE ROTATION IS NEGATIVE)

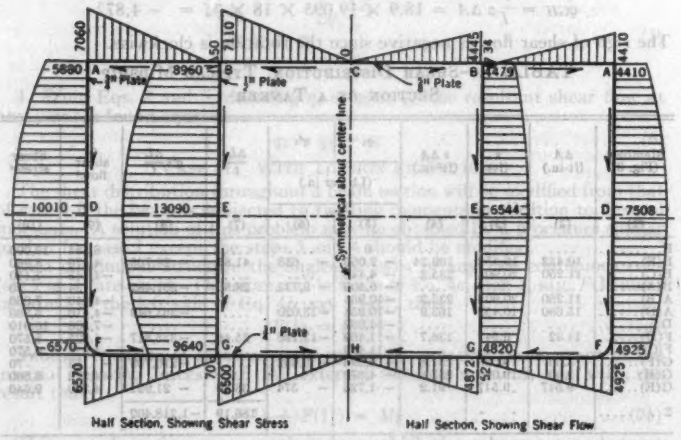


FIG. 4.—FINAL SHEAR STRESSES IN VESSELS WITH LONGITUDINAL BULKHEADS, IN POUNDS PER LINEAR INCH

By assuming a shear flow at point E in member BE the shear flow at point B in member BE may be obtained from Eq. 2*b*. Assuming the shear flow at point E equal to zero for convenience, the assumed shear flow is computed as in Cols. 3 and 4, Table 2. Similarly, by use of Eq. 2*b*, the assumed shear flow at each successive point may be computed. This shear flow is shown in Fig. 3.

The assumed shear flow will result in a twist of cell II. Since the angle of twist θ must be zero, a shear flow, acting around the cell, will be introduced as a correction to the assumed shear flow so that the resultant angle of twist will be zero. This can be done by Eq. 5, thus:

$$q_2 = \frac{1{,}218{,}402}{186.19} = 6{,}544 \text{ lb. per in.}$$

Applying this constant correction to the assumed shear flow around cell II (Col. 4, Table 2) gives the final shear flow as shown in Col. 5, Table 2, and in Fig. 4. The unit shear stress at any point (in pounds per square inch) is obtained by dividing the shear flow at the point by the thickness of the plate at the point:

$$v_E = \frac{q_E}{t_E} = \frac{6{,}544}{0.5} = 13{,}090$$

$$v_D = \frac{q_D}{t_D} = \frac{7{,}508}{0.75} = 10{,}010$$

If computed by the ordinary formula, the shear stress, $v = \frac{V Q}{I t}$ and $v_E =$ 11,250 lb. per sq. in.—or an error in this case on the unsafe side of 16%. The distribution of the shearing stresses is shown in the right half section of Fig. 4.

Summary.

The distribution of the shearing stress in a multiple-web beam may be found by the following steps:

1. Assume a statically consistent set of shearing stresses;
2. Find the angular distortion that results from these forces;
3. Introduce a set of shearing forces of such magnitude that the resultant angle of twist will be equal to that known to exist in the structure; and
4. Superpose stresses found in step 3 on those of step 1 to give the desired stress distribution.

BOOK REVIEW.

FLUID MECHANICS OF TURBO MACHINERY, By Geo. F. Wislicenus. McGraw-Hill Book Co., N. Y., 1947, Price $7.50.

This book is divided into five parts which develop the subject of Fluid Mechanics in logical sequence from the basic principles involved to their applications to all kinds of turbo-machinery. Included in the applications are water, gas and steam turbines, centrifugal and axial flow compressors, propellers and hydraulic couplings. After a brief introductory chapter there follows a section of three chapters dealing with dynamic similarity and Reynold's Law and their application to practical problems.

Part III constitutes the main portion of the book and develops the theory of Fluid Mechanics for one, two and three dimensional flow. Applications to axial and radial flow machinery are developed. The subjects of fluid friction and cavitation are given emphasis. While most of this section of the book is concerned with turbomachinery, the theory of propeller design is nevertheless adequately covered.

A distinctive feature of the book is the inclusion of a section on the mechanics of compressible fluids applied to turbomachinery. While this material is usually considered to be a subject to be treated in separate texts, it logically belongs in a book devoted to fluid mechanics. Consideration of the fluid mechanics of gas turbines and compressors is particularly valuable.

The fifth and last section of the book concerns the geometry of turbomachinery. Geometric constructions for making orthographic projections of various types of impellers, runners, etc., are given and these should prove of value to the engineer and draftsman. The inclusion of charts for the properties of air, shock polar diagrams for supersonic air flow and characteristic diagrams for axial flow turbomachinery add to the value of the book, as do the many excellent illustrations throughout the text.

This book presents in one volume the best American and

European thought on the subject of Fluid Mechanics of Turbomachinery and is recommended as a valuable text for the use of students, teachers and the engineering profession.

THE RADIO AMATEUR'S HANDBOOK (TWENTY-FIFTH EDITION—1948), By the Headquarters Staff of the American Radio Relay League.

The standard manual of amateur radio communication, revised and restyled in the light of current needs as a radio construction manual and training text for class or home study. 760 pages, including catalog section and 8-page topical index. 1712 illustrations, including 83 charts and tables, 79 basic formulas. Price, $2.00 in the United States, its possessions and Canada; $2.80 elsewhere.

In this edition of the Radio Amateur's Handbook the material on theory, principles and design has been revised completely and integrated with constructional data in the light of present-day needs. In addition to its usual annual revision, it contains several completely new chapters. Ultra-high-frequencies and the microwaves are treated separately and thoroughly. The problem of interference to broadcast reception is discussed in detail, from public relations to technical aspects. Emergency operation, assembling a station and ARRL operating organization are covered also in new chapters. In addition to several chapters devoted to high-frequency receivers, transmitters and antenna systems, the 1948 edition contains five chapters of material, fully illustrated in the usual manner, devoted exclusively to the very-high-frequencies, ultra-high-frequencies and microwaves. These chapters constitute in themselves a v.h.f.-u.h.f. textbook offering to the reader up-to-date information on construction, techniques and propagation factors in that field. The traditional preparedness of the radio amateur to furnish emergency communications to his community in time of natural disaster is fully discussed in the new chapter concerned with emergency operation. Information concerning the products of approved suppliers of amateur gear is conveniently available in the catalog section of THE RADIO AMATEUR'S HANDBOOK.

ASSOCIATION NOTES.

ADDRESSES.

PLEASE keep the Society advised as to your address. Your report is the sole means we have of ensuring delivery of your Journal and other communications.

PRIZE ESSAY CONTEST.

The Society's prize essay contest is attracting increasing attention. Ten entries were received in the contest which ended on March 1, 1948.

It is the Society's intention, through this contest, to foster the development of engineering by encouraging the writing of essays of a thought-provoking nature. For the most part, entries to date have been more in the nature of technical articles rather than essays.

The announcement of the contest for 1948 stated that prizes of $500.00, $300.00 and $200.00 might be awarded for papers considered by the Council to be worthy of such awards. Having in mind the purpose of the contest and, above all else, to maintain the very highest standards, the Council, upon thorough consideration, decided that no $500.00 prize would be awarded this year. The two articles which were considered to be the best of those submitted were awarded prizes of $300.00 and $200.00, respectively.

The Council selected a paper by Commander George C. Wells, U.S.N., entitled "The Importance of Controlled Humidity in Long Time Preservation" as the best submitted. It appears in this issue.

The second selection of the Council was submitted by Mr. Albert F. Bird. It is entitled "Photoprojection—New Challenge to Shipbuilding Tradition". This paper will be published in the August, 1948, issue.

Several other articles submitted are considered excellent and will be published in the next few issues of the JOURNAL.

CHANGE IN COUNCIL.

Because of retirement and removal from Washington, Captain D. R. Hull, U.S.N., submitted his resignation from the Council.

Pursuant to the By-Laws, the Council appointed Captain Rex L. Hicks, U.S.N., for the unexpired term of Captain Hull, ending December 31, 1948.

Society Lapel Button.

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold back-ground.

It is available to all members at fifty cents (50c) each.

Membership.

Attention of all non-member officers is especially invited at this time to the advantage and benefit to be attained by membership in this Society. The Society was organized in 1888—for the sole purpose of the advancement of Engineering. Its Journal, published quarterly, in February, May, August and November, now in its 60th year of continuous publication, has attained a foremost rank in its field and is a recognized authority throughout the engineering world. It is used generally by Naval officers and civilians alike as a reliable reference book. The Bureau of Ships considers the Journal of such value that it makes it available for the libraries of all Naval vessels and shore activities of any size. The presence of the Journal in your personal library cannot fail to be of great benefit to you, often making available information of great value which can be obtained from no other source.

It is particularly suggested that officers of the Naval Reserve, who are returning to civil life, should arrange for membership prior to detachment from active duty.

Annual dues are $5.00, which are probably lower than those of

any other technical Society. There is no initiation fee, nor any extra charge for the JOURNAL. This extremely low cost is possible only because the Society is bona fide operated solely for your benefit without any pecuniary profit to any one.

An application blank generally will be found at the back of each copy of the JOURNAL. However, if none is found, application by letter will be accepted.

It is suggested that Members will confer a real favor on non-members by bringing the Society to their attention.

MEMBERSHIP ADDITIONS.

The following have joined the Society since the publication of the February, 1948, JOURNAL:

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